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MODERN TURBINE PRACTICE AND WATER-POWER PLANTS

BY

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PREFACE.

Two years ago the writer published in *Engineering News* some articles on modern turbine practice, and the encouragement given by progressive engineers led him to rewrite these articles, to make large additions, and to include new subject-matter, which together form the contents of this volume.

The object of this book is to give such information in regard to modern turbines and their proper installation as is necessary to the hydraulic engineer in designing a water-power plant, and no attempt has been made to treat on the design of turbines, as to do this satisfactorily would require in itself a very large volume.

The writer has designed turbines both in America and in Europe, and has been connected in engineering capacities with water-power developments aggregating nearly 200,000 H.P., having been in charge of the hydraulic work during the planning and construction of some of the most important developments in Canada. The writer therefore had an excellent opportunity to study the subject from the point of view of both the turbine builder and the turbine user and thus became convinced of the necessity of thorough changes and improvements in the American turbine practice.

In the first part of this book the writer has shown the deficiencies of the present American turbine practice and pointed out the direction in which improvement is to be sought. It should not be inferred that what is said here regarding turbine design and construction is intended to depreciate the American and praise the European practice. Considered as a hydraulic motor, each type has a field of its own, where it should be used in preference to the

other. The present European practice has only been evolved during the last ten years and is yet in a transition state, so that changes and improvements are continually being made.

The American standard type of turbines has not been shown in the illustrations, as every hydraulic-power engineer is sufficiently familiar with it through the engineering press and the turbine catalogs.

On account of the growing importance of the steam-turbine and its close relation to the hydraulic turbine, the writer has included a chapter on this subject.

In the second part of this book will be found information and data about matters connected with turbine plants. These were either taken from the writer's personal experience or collected from recent volumes of the American and European engineering press.

Of course there are instances where the refinement in turbine design and construction here recommended does not pay, being either unnecessary or impracticable. For example, the backwoods sawmill, moving with the pioneer settler into newly opened territory, will do best with the roughest kind of a turbine plant.

The opinion of engineers in regard to many statements made in this book may vary from that of the writer, and no hard and fast rules can be laid down for water-power developments, as every case demands a careful and intelligent judgment, requiring greater experience on the part of the planning engineer than most other classes of construction work.

At the end of this book will be found Mr. Allan V. Garratt's paper on speed regulation of turbines, and the writer begs to thank here Mr. Garratt and the American Institute of Electrical Engineers for their kind permission to reprint this paper.

JOHN WOLF THURSO.

May, 1905.

ILLUSTRATIONS.

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G. Meissner, "Die Hydraulik und die hydraulischen Motoren," Vol. 2. Jena, 1896–97. Figs. 1 and 2.

The Canadian Engineer, Toronto, Ont. Figs. 15 to 18, 75 and 80.

Zeitschrift des Vereins deutscher Ingenieure, Berlin, Germany. Figs. 5 to 7, 19 to 23, 26 to 34, 36 to 44, 47 to 56, 66 to 69, and 76 to 79.

Schweizerische Bauzeitung, Zurich, Switzerland. Figs. 3 and 4, 8 and 9, 25, and 61 to 65.

FOOT-NOTES.

The names of some of the books and periodicals frequently referred to in the foot-notes, being inconveniently long, have been abbreviated as given below; and as in all cases the number of the page or illustration is mentioned, the edition of the books used is also given.

Frizell. Water-power.—Water-power: an Outline of the Development and Application of the Energy of Flowing Water. By Joseph P. Frizell. 2d edition. New York, 1901.

Meissner. Hydraulische Motoren.—Die Hydraulik und die hydraulischen Motoren. By G. Meissner. 2d edition. Jena, 1895–99.

Mueller. Francis-Turbinen.—Die Francis-Turbinen und die Entwicklung des modernen Turbinenbaues. By Wilhelm Mueller. 1st edition. Hannover, 1901.

Schweiz. Bauz.—Schweizerische Bauzeitung. Zurich.

Stodola. Steam Turbines.—Steam Turbines. By Dr. A. Stodola. Translated from the German by Dr. Louis C. Loewenstein. New York, 1905.

Taschenb. Huette.—Des Ingenieurs Taschenbuch. Herausgegeben vom Verein Huette. 17th edition. Berlin, 1899.

Wood. Turbines.—Turbines, Theoretical and Practical. By De Volson Wood. 2d edition. New York, 1896.

Zeitsch. d. V. deutsch. Ing.—Zeitschrift des Vereins deutscher Ingenieure. Berlin.

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NOMENCLATURE AND SYMBOLS FOR HYDRAULIC-POWER ENGINEERING.

In hydraulic-power engineering there exist so far no generally accepted terms, and not only are several different names given to most things, but what is worse, the same terms are often used with different meanings.

To start a movement towards uniformity, the writer would suggest here, for universal acceptance, the terms and meanings given below; and these terms have been used through this book, so that the reader, by referring to this nomenclature, may at once know exactly what is meant.

The terms here suggested are already used to a greater or less extent; they are not only simple and easily remembered, but most of them directly express their own meaning.

The characteristics and properties of the different classes of turbines will be found under "Classification of Turbines."

Uniform symbols have also been employed in all formulas appearing in this book, and the following system has been used: All heads are expressed by H, parts of heads by h; all speeds of water by c (celeritas or celerity); all speeds of runners by v (velocitas or velocity) and all efficiencies by η (eta). The single letters denote in all cases the most important or determining quantity or value, while all other quantities or values are denoted by the single letter as a base and an index-letter, the latter being the inital letter of a word which as nearly as possible expresses the particular meaning of the base letter. In cases where not readily understood, the word for which the index-letter stands is given in brackets.

While figures or dashes are more generally used for indices, the writer has adopted letters for this purpose, as being more easily remembered.

Action turbine.—For free-deviation turbine, such as the Girard, etc., say "action" turbine.

Axial turbine.—For "parallel-flow" turbine say "axial-flow" turbine, or axial turbine.

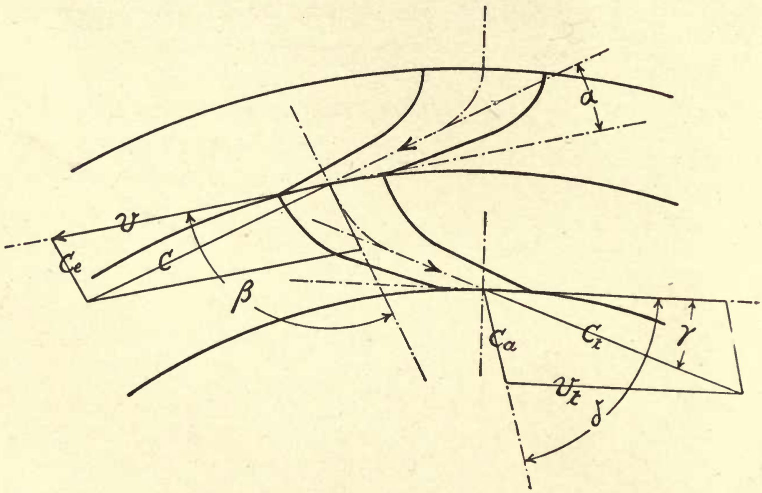


Diagram of Guide- and Runner-bucket, showing angles and velocities.

Bucket angles.—All angles are measured from the tangent at the point where the direction in question intersects the circumference. (See figure.)

α = terminal angle of guide-bucket.

β = initial angle of runner-bucket.

γ = terminal angle of runner-bucket.

δ = actual angle of the jet of water at exit from runner-bucket.

Bucket circle. — The imaginary circle running approximately through the centre of the radial dimension of the buckets of impulse turbines and to which the centre line of the jet of water is a tangent should be called "bucket circle." This circle is somewhat similar, in its importance and use, to the pitch line of a gear.

Clearance.—The clear space between the guide-ring and the runner should be called the "clearance."

Dip of draft-tube.—For the vertical distance to which a draft-tube reaches below the surface of the tailwater, say dip of draft-tube.

Draft-tee.—For "draft-chest" or "camelback" say "draft-tee."

Efficiencies.—As there are four different efficiencies to be considered in connection with water-power developments, specifications and contracts should always state which efficiency is meant; also, where the tests are to be made.

η_h = hydraulic efficiency. This is the ratio of the power actually developed by the water in flowing through the guides, runner, and draft-tube, if a draft-tube is employed, to the theoretical power due to the effective head. The hydraulic losses are the friction and shocks of the water in guides, runner, and draft-tube, and the velocity head, corresponding to the absolute velocity of the water at exit from runner-buckets, or to the velocity of the water at exit from the draft-tube, if such is used. Reports on efficiency tests should always state if a draft-tube was employed.

η_m = mechanical efficiency. This is the ratio of the power delivered at the end of the turbine-shaft to the power actually developed by the water in flowing through the guides, runner, and draft-tube, if a draft-tube is used. The mechanical losses are the loss of water through the clearance, the friction of the runner in the air or in the water, according to the location of the turbine, and the friction of the shaft in its journal- and step-bearings. If the turbine is enclosed in a case, the friction of the shaft in the stuffing-boxes and the friction of the water in the case itself are additional losses.

η = turbine efficiency. This is the efficiency of the turbine as a whole, or the ratio of the power delivered at the end of the turbine-shaft to the theoretical power due to the effective head. The turbine efficiency is equal to the product of the hydraulic and mechanical efficiencies, and takes into consideration both the hydraulic and the mechanical losses.

η_t = total efficiency, or efficiency of the entire plant. This is

the ratio of the power delivered at the end of the turbine-shaft to the theoretical power due to total head utilized in the development. Besides the hydraulic and mechanical losses, all losses in the headrace, penstock, tailrace, etc., are here taken into consideration.

External-feed turbine.—A turbine having the guide-ring outside the runner and discharging inward should be called "external-feed turbine."

Full turbine.—A turbine having guide-buckets around its whole circumference is a "full turbine."

Gate.—The term gate or gates should always mean the speed regulating gate or gates of a turbine.

Gate opening and discharge.—The area left open or clear by the regulating gate or gates for the passage of the water should be called gate-opening. At present the term gate-opening or gate is nearly always used to mean the amount of water flowing through the gate-opening, but this amount should be designated by discharge; for example, instead of saying: "This turbine with five-eighths gate-opening gave an efficiency," etc., should be said: "This turbine with five-eighths discharge gave an efficiency," etc., whenever the discharge is meant.

Heads of water, in feet.

H_t = total head utilized in a development.

H = effective head, that is, the head available at the turbine, equal to the total head less the losses in headrace, penstock, tailrace, etc.

h_d = draft-head, that is, the part of the head which is utilized by means of a draft-tube. The height of the draft-head acting on a radial- or parallel-flow turbine on horizontal shaft is the vertical distance from centre of shaft to the tailwater level, while the draft-head of a radial turbine on vertical shaft is the vertical distance from the level of the centre of the guide-bucket discharge-openings to the level of the tailwater. The draft-head of an impulse turbine or an action turbine with free deviation is, of course, the vertical distance from the water level in the turbine-case or draft-tube to the level of the tailwater.

h_p = pressure-head, that is, the part of the head which is above the turbine and terminates at the point where the draft-head commences, as explained under Draft-head.

h = velocity-head, corresponding to the velocity of the water at exit from guide-buckets of reaction turbines. For action turbines this velocity-head is H.

h_r = head remaining as pressure of the water at exit from guide-buckets of reaction turbines. ($H = h + h_r$.)

h_a = velocity-head, corresponding to the absolute velocity of the water at exit from runner-buckets.

h_f = velocity-head, corresponding to the velocity of the water at exit from draft-tube. (f = final.)

Impulse turbines.—For Pelton or impulse wheel say "impulse turbine." While Mr. Lester A. Pelton, who claims to have built such turbines as early as 1864, appears to have been the inventor of this type, yet there are now so many variations of such turbines that a more general name should be used for same.

Inflow reaction turbine.—For Francis turbine say "radial inward-flow reaction" turbine, or simply "inflow reaction" turbine, as this type is usually understood by the name Francis turbine. While it is not likely that the term Francis turbine will go out of use, the term inflow reaction turbine should always be used where an exact expression is of importance, as in contracts, for the reason that Mr. Francis also designed outflow reaction turbines.

Internal-feed turbine.—A turbine having the guide-ring inside of the runner and discharging outward should be called "internal-feed" turbine.

Length of draft-tube.—The distance from the centre of the cross-sectional area at the top end of a draft-tube to the centre of the area at the discharge end, measured along the centre line, is called the length of a draft-tube, and should also include such parts of the turbine-case or draft tee or elbow as have approximately the same area and shape as the draft-tube. However, when the term is used in connection with the effect of a flaring draft-tube on the efficiency, etc., only that part of the length is to be considered in which the cross-sectional

area is gradually and continually increasing toward the discharge end.

Limit turbines.—Haenel turbines, that are action turbines without free deviation, should be called "limit" turbines, because the jet of water in the runner-bucket is limited by the back of the next vane, and also because such turbines form the limit between action and reaction turbines.

Miscellaneous symbols.

Q = quantity of water in cubic feet per second. The miner's inch, which is measured in a number of different ways and therefore is no exact unit, should be abandoned altogether.

P = pressure or reaction, in pounds, produced by a jet of water.

W = width or clear space between crowns of runner at initial or entrance rim.

W_t = width or clear space between crowns of runner at terminal or exit rim.

ε = angle through which a jet of water is deflected by the runner-buckets.

n = revolutions per minute.

g = acceleration of gravity $= 32.16$ ft.; $\sqrt{2g} = 8.02$.

Outflow reaction turbine.—For Fourneyron or Boyden turbine say "radial outward-flow reaction" turbine, or else simply "outflow reaction" turbine. It may here be stated that the first turbines of Mr. Fourneyron, designed in 1827, were outflow action and his later ones outflow reaction turbines, and the latter turbines are usually understood by the name Fourneyron turbine.

Partial turbine.—A turbine having guide-buckets only on part of its circumference, arranged in one, two, or more groups, is often known as segmental-feed turbine, but should be called "partial-feed turbine," or simply "partial turbine."

Penstock.—For "feeder-pipe" or "water-feeder" say "penstock."

Penstock speed.—For speed of water in the cylindrical or main part of the penstock say "penstock speed."

Power-water.—The water available under the head utilized should be called "power-water," corresponding in meaning to the term live steam.

Reaction turbine.—A turbine working with reaction, as the Jonval, Francis, etc., should be called "reaction" turbine.

Relay.—The motor furnishing the power for actuating the regulating-gates of a turbine, and usually controlled by a speed-governor, should be called a relay. In Europe the term servo-motor is used for such auxiliary machines.

Return.—To prevent overgoverning or racing, a device called the "return" is used with modern turbine-governors, to arrest the motion of the regulating-gates before moving beyond the required position.

Ridge.—The radial edge dividing the buckets of impulse turbines into halves should be called the "ridge." The common name for it at present is wedge or splitter, but in some designs of impulse turbines, having double water-jets, the ridge is not used for wedging into or splitting the water-jet.

Right- and left-hand turbine.—When looking at a radial inflow turbine or a vortex turbine in the direction of the shaft and facing the end opposite to the discharge end, that is, looking in the direction in which the water leaves the turbine, the turbine is a right-hand one if it turns with the sun or in the direction of the hands of a watch, and a left-hand one if it turns in the opposite direction.

Runner diameters, in feet.

D = diameter of initial or entrance rim of runner. For impulse-turbines D is the diameter of the bucket circle, or twice the vertical distance from the centre line of the jet of water, or the prolonged centre line of the nozzle, to the centre of the runner-shaft.

D_t = diameter of terminal or exit rim of runner.

Runner speeds, in feet per second.

v = speed of runner at initial or entrance rim. For impulse turbines v is the speed at the bucket circle.

v_t = speed of runner at terminal or exit rim.

Speed factor.—The ratio of the speed of the initial rim of a runner to the theoretical discharge velocity of water $(\sqrt{2gH})$ under the effective head used by the turbine in question should be called "speed factor." This speed factor will at once show if the turbine is an action, limit, or reaction turbine,

and if the latter, with what amount of reaction the turbine is working.

Speed variation.—The performance of a governor is usually given by stating the limits of variation within which the governor will keep the speed of a turbine; but as there are two ways of expressing this, it should always be stated what variation is meant. Thus if the variation between a maximum and a minimum speed is meant, it should be called "total speed variation," and if the variation either way from the normal speed, above and below, is meant, it should be called "speed variation from the normal."

Stop-gate.—This is a gate or gate-valve in the penstock or penstock nozzle located near the turbine, or between penstock or nozzle and turbine-case, or in the turbine-case itself, used for shutting down the turbine and corresponding to the stop-valve of a steam-engine.

Stop-logs.—These are wooden or steel beams which form, when placed in the slides provided for them, a gate or coffer-dam, and are used to keep out the water from a headrace or tail-race, turbine-chamber, etc., to admit of inspection or repairs.

Tailwater.—The water having descended either through the turbines or over the falls, should be called "tailwater," corresponding in meaning to the term exhaust steam.

Turbine.—For water-wheel say "turbine" whenever a turbine is meant. For one, two, three, four, five, or six turbines on one shaft say single, double, triple, quadruple, quintuple, or sextuple turbine. The terms horizontal or vertical turbine should always mean a turbine on a horizontal or vertical shaft, and never one revolving in a horizontal or vertical plane, as it is now sometimes understood.

Turbine-chamber and flume.—For open flume say "turbine-chamber" whenever a turbine-chamber is meant, leaving the term "flume" to mean a water-conductor only, built of wood, steel, or masonry, and carrying water not under pressure.

Vanes.—The partitions dividing the space between the crowns of the guide and runner into the buckets and which, by their shape and angularity, determine the working of a turbine, are usually called "vanes." The side against which the water

exerts its pressure, either by its deflection or by deflection and reaction, and which side is nearly always concave, is the "face," the other side the "back," of the vane.

Vortex turbines.—Turbines of the American type, in which the water when entering the runner-buckets, flows radially inward, then axially and leaves the bucket at a slant, between the axial and radial outward-flow direction, are usually called "vortex" turbines.

Water-speeds, in feet per second,

c = speed of water at exit from guide-buckets, and at the same time the absolute entrance speed of water in runner-buckets.

c_e = initial or entrance speed of water in runner-buckets, relative to the buckets.

c_t = terminal speed of water in runner-buckets, relative to the buckets.

c_a = absolute speed of water, that is, the speed relative to a stationary object, at exit from runner-buckets.

c_b = initial speed of water in draft-tube, that is, the speed of water, while entering the upper end of the draft-tube. (b = beginning.)

c_f = speed of water at exit from lower end of draft-tube. (f = final.) This speed, being of great importance in connection with the efficiency and proper working of a turbine, should be called "draft-tube speed."

Wing-gate.—For butterfly-gate say "wing-gate."

DRAWINGS.

It would also be well to adopt some uniform names for sections and views of turbines, and the writer would suggest the following:

For horizontal turbines: A vertical section through the centre line of or parallel to the shaft is a "longitudinal" section, and a vertical section at right angles to the shaft is a "cross"-section. A view looking at right angles to the shaft is a "side" elevation, and looking in the direction of the shaft an "end" elevation. The names plan and horizontal section cannot well be misapplied.

For vertical turbines: A section through the centre line of or parallel to the shaft is a "vertical" section, and a section at right angles to the shaft is a "horizontal" section. A view looking at right angles to the shaft is an "elevation," and looking in the direction of the shaft a "plan."

PART I.

MODERN TURBINE PRACTICE.

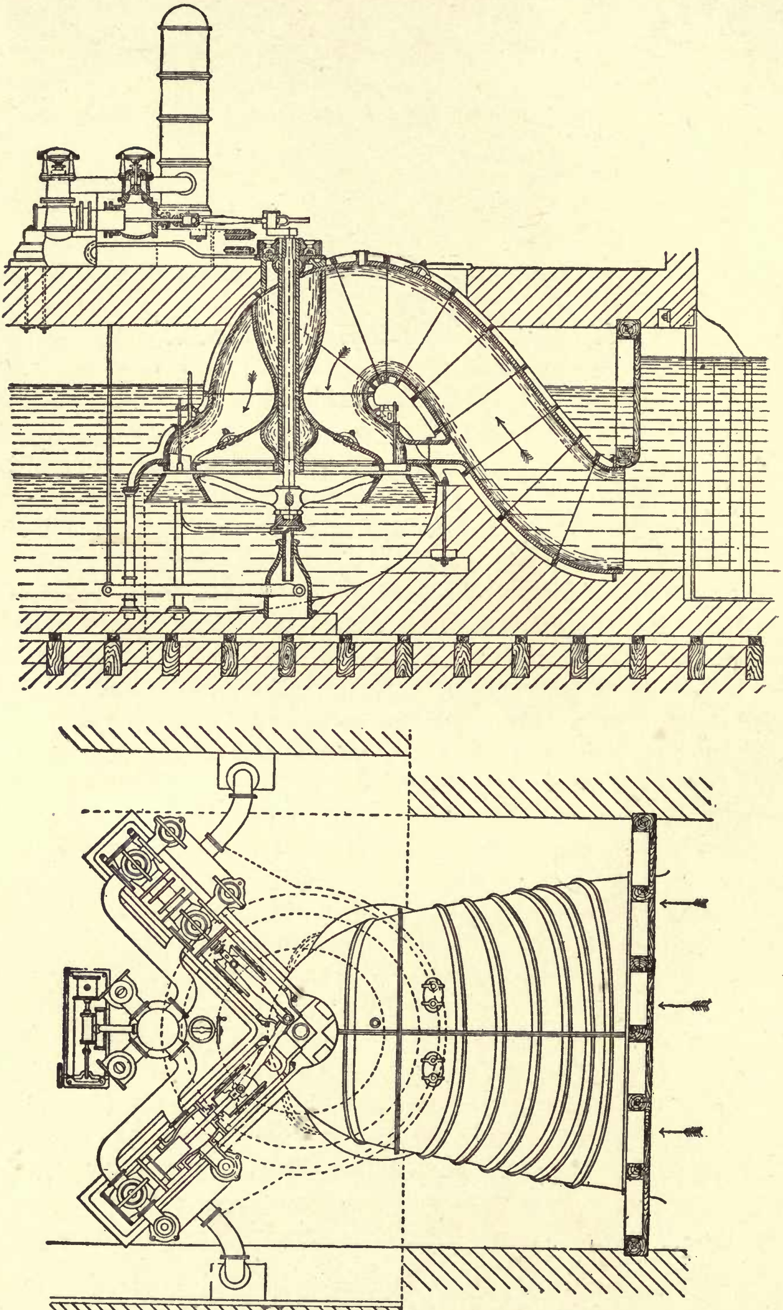
CHAPTER I.

TURBINE PRACTICE IN EUROPE.

Turbine Development in Europe.—The turbine may be called a theoretical invention, as it is one of the few machines which have been invented as the direct outcome of mathematical investigation, and not by experiment, like the steam-engine and others, and its further evolution in Europe has always been based on theory. In America, however, turbine construction has been developed almost solely by experiment.

Although most of the early turbines were of the radial-flow type, the axial-flow turbine soon became the standard type of European builders, and remained so until a few years ago, while the radial-flow turbine was rarely used, except for partial turbines working under high heads.

An interesting example of the axial-flow type is the turbine at the water-works of Geneva, Switzerland, shown in Figs. 1 and 2. This turbine drives two plunger-pumps with vertical crank-shaft, and develops from 50 to 100 H.P. under a head varying between 16½ and 35½ ins., running from 9 to 16 revolutions per minute, and using from 530 to 706 cu. ft. of water per second. The siphon-feeder was employed to save tailrace excavation, which is in hard rock, as without the siphon the turbine would have to be set much lower. The efficiency is naturally low for such small



Figs. 1 and 2.—50-100-H.P. Turbine at the Water-works of Geneva, Switzerland. Built by Girard & Callon, Paris, France.

heads, as the mechanical losses are great in proportion to the power developed. This turbine and pumps, with a head of 16½ ins., have a combined efficiency of 48%, or say 58% for the turbine alone.[1]

This turbine was designed by Mr. Girard and therefore is an action turbine. At present vertical-inflow reaction turbines are always used for very low heads.

Axial-flow turbines are simple and cheap, but no satisfactory way of regulating their speed could be found, and with the advent of electrical transmission, requiring close regulation and higher speeds, builders cast about for a new type of turbine. A great variety of new designs were then brought forward, and while as yet none have been definitely and generally adopted, some of the designs used by the most progressive builders are certain to be soon recognized as the coming standard types. These designs are:

For low heads, say up to 20 ft.: Radial inward-flow reaction turbines with vertical shafts, the foundation masonry usually forming the turbine-case, draft-tubes being frequently employed. (See Figs. 19, 20, 22, and 23.)

For medium heads, say from 20 to 300 ft.: Radial inward-flow reaction turbines with horizontal shafts and concentric or spiral cast-iron cases with draft-tubes. (See Figs. 26 to 41.)

For high heads, say above 300 ft.: Radial outward-flow, full- or partial-action turbines, with horizontal shafts and cast- or wrought-iron cases, frequently with draft-tube. (See Figs. 42 to 48.)

A modified impulse turbine has of late been gaining in favor, and it is most likely that the type ultimately adopted for high heads will be a combination of the best points of the partial-action turbine and the impulse turbine. (See Figs. 49 to 56.)

As the axial-flow turbine has practically been abandoned, the following refers to radial-flow turbines only.

Of all European countries, Switzerland, having many excellent water-powers and producing no coal whatever, ranks highest in progressive turbine-building, being closely followed by

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[1] Meissner. Hydraulische Motoren, vol. 2, p. 588.

Germany, Austria, and Italy. France, although the native country of the turbine, and rich in water-powers, has advanced little during the last ten years, except that some builders are closely copying American turbines, such as the Hercules. Most of the turbines installed in important French plants are supplied by Switzerland.

Mr. Prasil, professor of the Polytechnikum in Zurich and one of the prize-judges for the turbines at the Paris exhibition in 1900, says in his report:[1] "Modern French turbine-building at the exhibition was characteristically shown by the Hercules type. Besides this, only old types were to be seen. Little attention seems to have been paid by French designers to the progress in turbine-building made in Central Europe, especially to the solution of the problem of regulation."

However, French builders have recognized the fact that the same pattern of turbine can work economically only under a certain range of heads. For example, one of the largest manufacturers of turbines of the Hercules type[2] offers four series of pattern, to suit heads as follows:[3] Series 1, 3.25 to 10 ft.; Series 2, 10 to 25 ft.; Series 3, 25 to 40 ft.; Series 4 (special), 40 to 108 ft.

The application of the laws of hydraulics to the action of water in simple radial- or axial-flow turbines being well understood since early in the nineteenth century, the highest possible hydraulic efficiencies could be calculated for the different types of turbines, and multiplying these values, which vary between 0.80 and 0.90 for the usual designs, by the highest possible mechanical efficiency of the turbine under consideration, the limit of the possible total efficiency is found. This limit, which may be considered to lie between 0.75 and 0.85 for practical working types, has been reached, or very closely approached, with one half to full discharge at least, by the regular turbines, such as the best European builders have turned out during the last twenty years or more. Their present efforts are, therefore, to make the turbine a complete and self-contained machine, to improve the details, such as step- and journal-bearings, regulating-gates and their

[1] Schweiz. Bauz., Feb. 16, 1901, p. 74.

[2] Singrün Frères, Epinal, France.

[3] Mueller. Francis-Turbinen, p. 273.

rigging, oiling of bearings under water, etc., to raise the speed for turbines working under low heads, and, as has been the endeavor for a long time, to improve the efficiency of reaction turbines when running with only a fraction of their full discharge.

Builders also have realized the expense involved in making separate patterns for every turbine built, and all the large manufacturers have now lists of standard sizes, furnishing specially designed turbines only when the occasion requires it.

Present Turbine Practice in Europe.—The principal features only of the present European practice are here dealt with, while examples of European turbines will be described later on.

The question of efficiency with part loads has been so well solved for action turbines that further improvements are not to be expected. Efficiencies of 70 % for 0.2 discharge to 80% for full discharge are the common practice, and these results have been reached by very simple means. With full turbines an ordinary cylinder gate is used, as shown in Figs. 42 to 44, reducing the outflow-openings of all guide-buckets equally, in proportion to the decrease in power developed, while with partial turbines a slide is employed, as shown in Figs. 45 and 46, which closes the outflow-opening of one guide-bucket after the other as the load on the turbine is reduced.

Greater difficulties, however, are met with when it is attempted to regulate reaction turbines so as to show high efficiencies with part loads. The simplest means are in this case the most wasteful, and have practically been abandoned, but may be mentioned here, viz., the throttling of the water in the inlet-pipe or in the draft-tube by wing-gates (see Figs. 57 and 58), or at the lower end of the draft-tube by a cylinder gate, closing the annular space between the draft-tube and the bottom of the turbine-pit. For high heads and very long penstocks a by-pass was used in connection with the throttle-valve.

The three methods of regulating reaction turbines most widely used in Europe at the present time are as follows:

1. *The cylinder gate.* The width of the guide- and runner-buckets, or the distance between the crowns, is divided into two or more spaces by additional crowns, thus forming two or more turbines, which are regulated by one common cylinder gate.

Thus a triple—or three-story—turbine, when working with one-third or two-thirds gate-opening, has one or two turbines working with full gate and full gate efficiency, while the two turbines or one remaining are shut off entirely.

The cylinder gate in nearly all cases works between the guides and runner. As an example of recent design may be mentioned the 700-H.P. horizontal-inflow turbine built by Siccard, Pictet & Co., Geneva, Switzerland, for a power plant in southern France.[1] These turbines have a width of buckets of nearly 30 ins., divided into five stories, all regulated by a single gate. The 5500-H.P. turbines in power-house No. 1 of the Niagara Falls Power Co., at Niagara Falls, N. Y.,[2] are examples of three-story turbines with the cylinder gate on the discharge side of the runner. Turbines with a width of buckets not sufficient to permit the use of additional crowns should not be regulated by cylinder gates, as the economy is poor with part loads.

2. *The Register Gate.* A part of each of the vanes which form the guide-buckets is separate from the rest of the vane, being attached at each crown to a movable ring, so that by rotating these rings the size of each of the clear openings between the vanes can be altered simultaneously in accordance with alterations in the load of the turbine. The movable part of the vanes may be either at the entrance or at the discharge side of the guide-buckets. The former plan is now rarely used, as the shape of the bucket is too much distorted when the gate is partly closed.[3] With the gate at the discharge side the shape of the bucket is much better maintained, and such an arrangement is used by many builders. Figs. 28 and 29 show such a gate. A great improvement over the ordinary register gate is the Zodel gate. Here also the discharge end of the vanes is movable, but a steel plate is bolted to the back of the stationary part of each vane, thus retaining the proper shape of at least one side of the buckets at all positions of the gate. In Figs. 3 and 4 the Zodel gate is shown in three positions, viz., fully and half open and closed. When fully closed, the movable part of the vanes forces the steel

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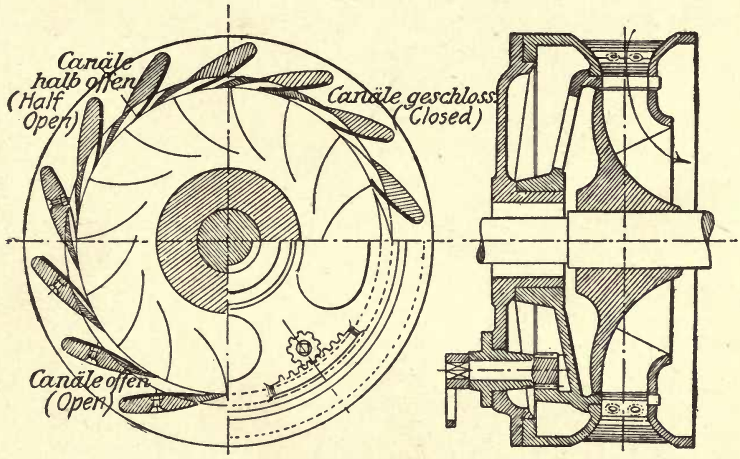
[1] Zeitsch. d. V. deutsch. Ing., Nov. 16, 1901, p. 1633.

[2] Wood. Turbines, Fig. 57.

[3] Schweiz. Bauz., Feb. 16, 1901, p. 72, Fig. 14.

plates slightly outward, and these steel plates, acting as springs, are thus insuring a tight joint. The turbines shown in Figs. 25, 30, 31, 36, and 37 are regulated by Zodel gates.

3. *The Wicket Gate.* The whole guide-vane swings on pivots so located as to balance the vane as nearly as it is possible in every position. Here also all the vanes move and thus alter the size of the discharge-openings of the buckets simultaneously. Such gates maintain the correct shape of the guide-buckets at part gate opening better than any other gate. The most widely



FIGS. 3 and 4.—Zodel Register Gate. Gate shown in open, half-closed, and closed position.

used design is the Fink gate, which is shown in connection with the turbines illustrated in Figs. 26 and 27 and 32 to 34.

A regulation which may be called either a register or a wicket gate is used for the turbine shown in Figs. 40 and 41, and is also employed for the large turbines at Beznau. (Fig. 19.) This gate has part of each vane movable, which part is located on the entrance of the buckets and both swings and moves inward at the same time, the inner ends of the movable part of one vane and the stationary part of the next vane meeting when the gate is closed. This gate also maintains a good bucket shape, except for very small gate-openings.

The ideal regulation of reaction turbines would be the simultaneous altering of the size of all the clear openings of both the guide- and the runner-buckets, in accordance with alterations of the load, and without changing the curvature or shape of the buckets. This could be attained by making one crown of each, the guide and the runner, movable, so that the size of the bucket-openings could be varied by varying the distance between the crowns, but all attempts to carry this out in practice have so far failed.

However, even with the imperfect gate arrangements, European builders now commonly realize the following efficiencies with reaction turbines:[1]

Discharge	0.2	0.3	0.4	0.5	0.6
Efficiency	60%	70%	76%	79%	80%
Discharge	0.7	0.8	0.9	1.0	
Efficiency	81%	82%	81%	80%	

It should be noted here that the highest efficiency of reaction turbines is usually found to be at about 0.8 of the full gate discharge, although such turbines may be designed to give their maximum efficiency at any desired discharge.

Action turbines must, for good efficiency, work in the air; that is, above and clear of the tailwater. Therefore, to enjoy the advantages of a draft-tube with this class of turbines, they are now furnished with an air admission-valve, which automatically regulates the water level in the turbine-case or draft-tube, so as to keep it just below and clear of the turbine-runner. (See Figs. 42 to 46.)

Outward-flow turbines can either be designed to have no end thrust whatever, and turbines on horizontal shafts are usually arranged in this manner (see Figs. 42 to 48), or to have any desired amount of end thrust, up to the limit set by the head of the water and the size of the turbine. Such end thrust is often employed to support the weight of the rotating parts of turbines on vertical shafts. As an example may be mentioned the 5500-H.P. turbines at Niagara Falls, N. Y., where the lower turbine of each pair has no end thrust, while the upper one has an upward thrust,

[1] Zeitsch. d. V. deutsch. Ing., Nov. 22, 1902, p. 1790

which carries the runners, shaft, and rotating field, weighing together about 80 tons. This large weight is practically floating on the water, and the easy working of these turbines is well illustrated by the fact that without load they will run at a speed of 40 revolutions per minute, with the turbine-gates entirely closed and only using the water which leaks through the small clearance between the runners and the gates, which clearance is only $\frac{1}{16}$ in. If it is desired to stop the turbines without closing the head-gates, a brake has to be applied to bring the turbines to rest.[1]

Single inward-flow reaction turbines, and to a much smaller degree certain types of inward-flow action turbines, present great difficulties in taking care of the end thrust, especially troublesome in connection with high heads. This difficulty has been successfully overcome by European builders by two different means, used according to the conditions. These means are the following:

1. *The Thrust Piston.* This arrangement is principally used for vertical-inflow turbines and takes the thrust of both the action of the water and the weight of the rotating parts. As a conspicuous example, one of the 5500-H.P. turbines for power-house No. 2 of the Niagara Falls Power Co. is shown in Fig. 5.[2]

This is a single vertical inward-flow reaction turbine in an approximately spherical case, the turbine being at the top, and the shaft is extended downward into the draft-tube and carries the thrust piston at its lower end, at the apex of the Y branches of the draft-tube. The piston is grooved to reduce the leakage, and the cylinder in which the piston rotates has a renewable lining. The pressure water acting at the lower side of the piston is taken directly from the headrace, screened to keep out dirt and gritty matter, and carried to the cylinder by an independent pipe. The upper side of the piston is subjected to the partial vacuum or suction in the draft-tube. The rotating parts,

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[1] Coleman Sellers. The Power Station at Niagara Falls. Trans. Am. Soc. M. E., 1898. Also reprint in Engineering, London, Jan. 20, p. 91, Jan. 27, p. 128, and Feb. 3, p. 160, 1899.

[2] Zeitsch. d. V. deutsch. Ing., Aug. 31, 1901, p. 1239; also an abstract in Engineering Record, Nov. 23, 1901, p. 500.

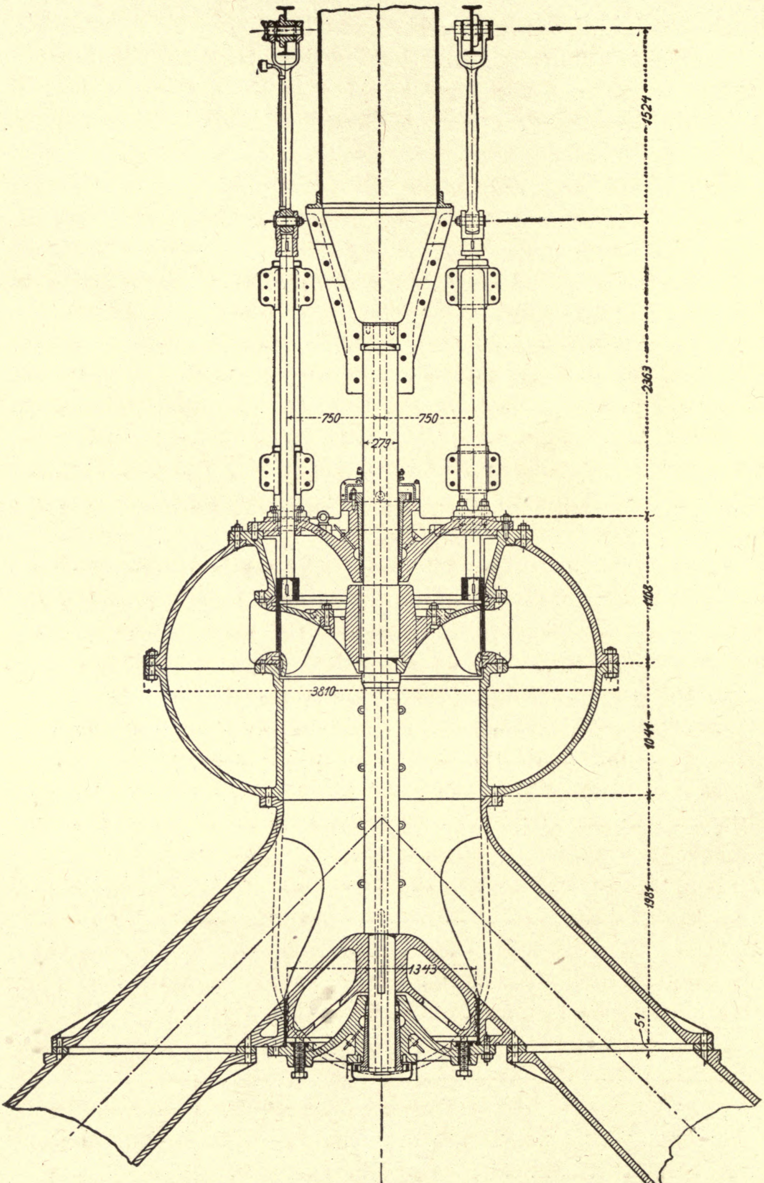

FIG. 5.—5500-H.P. Turbine for Power-house No. 2. of the Niagara Falls Power Co., Niagara Falls, N. Y. Built by Escher, Wyss & Co., Zurich, Switzerland.

viz., the runner, shaft, and revolving field of the dynamo, have a total weight of 71 tons, and of this weight the piston carries from 66 tons with full gate-opening to 77 tons with the gate nearly closed, the difference between weight and upward thrust being taken by a collar-bearing above.

The variation in effective upward thrust with different gate-openings is principally due to the variation in the pressure above the turbine-runner and the variation in the amount of vacuum in the draft-tube. With full gate-opening, water under pressure will be forced through the clearance into the space above the runner, while the fan-wheel action of the runner tends to force the water outward, thus creating a pressure above the runner. With the gate nearly closed, practically the same vacuum will exist above the runner as in the draft-tube. This is owing to the fact that with very small gate-openings the water in a reaction turbine works with action only, as will be explained when considering the effect of speed-regulating gates.

With the amount of water passed by the gate-opening decreasing, the velocity with which the water leaves the lower end of the draft-tube decreases in the same proportion and the vacuum therefore increases, as will be explained under draft-tubes.

The turbine seen in the power-house cross-section, Fig. 20, is also provided with a thrust-piston, carrying 22 tons of the weight of the revolving parts, but here the turbine is at the bottom of the case—in other words, hangs on the shaft—while the thrust-piston is at the top and the water-pressure in the case acts directly on its lower surface. The water leaking past the piston into the space above is led into the draft-tube by a waste-pipe, as shown in Fig. 20, thus reducing the pressure above the piston to that in the draft-tube. The piston area may be made larger than required for balancing the thrust and by closing the valve in above-mentioned waste-pipe more or less, the pressure above the piston is increased or decreased, and the remaining total upward pressure can thus be closely regulated.

2. *The Thrust-chamber.* This arrangement is principally used to take the end thrust of horizontal inflow turbines and consists of an annular chamber, formed by the cast-iron turbine-case and open towards the runner, which revolves in front of it, a

shown in Figs. 30 and 34. The water-pressure in the chamber is supplied by a pipe, the pressure being regulated by a valve in that pipe. The arrangement of the pipe connection is in accordance with the class of turbine employed and therefore two different cases have to be considered.

(*a*) *The Inflow Reaction Turbine.* Here the water is under pressure while passing the clearance and will therefore leak into the space back of the runner, that is, between the runner-disk and the head of the turbine-case and into the thrust-chamber. Opposed to this pressure is, at the back of the runner-disk, the fan-wheel action of the latter and the force required to press the water through the openings in the disk. In the thrust-chamber pressure is opposed by the fan-wheel action of the runner and the force required to press the water through the clearance space at the inner edge of the chamber. As a rule, the total pressure back of the runner-disk will exceed the total pressure due to the leakage into the chamber, and therefore pressure water has to be admitted to the latter, yet owing to differences in clearance-spaces, in shape or roughness of the back and chamber side of the runner and other minor causes, the total pressure in the thrust-chamber may rise above the total pressure at the back of the runner, and to relieve this pressure the chamber is also connected to the draft-tube, as shown in Fig. 30.

(*b*) *The Inflow Action Turbine with Limited Buckets.* Here the water is not under pressure while passing the clearance and no water will be forced into the thrust-chamber, but the fan-wheel action of the runner may draw water into the space back of the runner and, throwing it outward, create a pressure, which is balanced by admitting pressure water to the thrust-chamber. (See Fig. 34.)

By means of the valve in the pipe connecting the thrust-chamber with the pressure water and draft-tube or the pressure water only, the end thrust can be so regulated that the shaft will press against the step-bearing with just enough force to prevent any end motion.

The valves for regulating the pressure in the space back of the runner and back of the thrust-piston or in the thrust-chamber are operated by hand, but it may prove of advantage to use auto-

matic pressure-regulating valves instead, maintaining a fixed difference in pressure in these two spaces.

It has been attempted to regulate the pressure in the thrust-chamber and back of the runner by making the width of the clearances between runner and guides and inner edge of thrust-chamber and between runner and guides or turbine-case in proper proportion, but as these clearances should not be wider than from $\frac{1}{32}$ to $\frac{1}{16}$ in. and will wear larger and wear unequally, this plan has not met with any success.

For turbines having runners of such a shape as not to permit the use of the thrust-chamber, as for example the American or vortex turbine (see Fig. 9, also Fig. 26), the thrust-piston can always be used instead, having the runner on one end and the piston on the other end of the case.

The use of wood with water lubrication for bearings and steps located under water has been abandoned altogether by European manufacturers, and metal bearings and steps with forced oil lubrication are employed instead, using a pressure and return pipe for circulating the oil. The pipe connections to the bearing above the runner and the one below the thrust-piston are shown in the illustration of the Niagara Falls turbines, Fig. 5, and similar connections are shown in the illustration of a horizontal double turbine, Fig. 25.

In general it may be said that European builders have brought out and perfected a number of turbine types, which have enabled them to meet any reasonable conditions, and to predetermine and guarantee a high efficiency in all cases; for the small turbine of a few horse-powers as well as for those of 5500 H.P. at Niagara Falls, and for the head of 16½ ins. used at Geneva, Switzerland, as well as for the highest heads, such as those used at Vernayaz, also in Switzerland, where the six turbines at the electric power station are developing 1000 H.P. each under a head of 1640 ft. One of these turbines is shown in Figs. 47 and 48.

Turbine-pumps.—The reversed turbine or turbine centrifugal pump, like the turbine itself, was evolved by theoretical investigations.

Turbine designers reasoned correctly that when a turbine with a certain head and quantity of water gives a certain horse-

power and number of revolutions, then a similar turbine, being driven from some outside power at the same speed but in opposite direction, will act as a pump, elevating the same quantity of water to the same height and requiring the same horse-power as was developed by the machine when used as a turbine. Of course, allowance has to be made in practice for the losses in both the turbine and the pump.

Such turbine-pumps are of recent date only, yet some remarkable results have been obtained with them.

While theoretically all turbines could be reversed, the radial inward-flow type was naturally chosen, and the buckets and other water passages were somewhat modified in shape, as the direction in which the water flows through the pump is opposite to that in which it flows through the turbine The turbine-runner acts as the fan-wheel of the pump and the guide-buckets convert the speed of the water imparted to it by the fan-wheel into pressure, and it is this application of the guide-buckets which constitutes the great improvement and permits the turbine-pump not only to work economically against heads considerably higher than could ever be attempted with the ordinary centrifugal pump, but also to give a much better efficiency than can be obtained with the ordinary pump even under low heads.

For comparison it should be stated here that the ordinary centrifugal pump shows a maximum efficiency with a lift of about 17 ft., this efficiency being between 50% and 65%, according to the shape of the fan-wheel and case and the size of the pump.

As another case of the reversibility of turbine and centrifugal pump may be mentioned the reversed centrifugal pump, that is, the radial inward-flow turbine without guide-buckets, which has repeatedly been brought out by manufacturers,[1] but, like the ordinary centrifugal pump, has always given such poor efficiencies even with low heads and a rapid further decrease in efficiency with increasing heads, the same as the centrifugal pump, that such turbines have never been used to any extent.

A number of turbine-pumps, shown in Figs. 6 and 7, were

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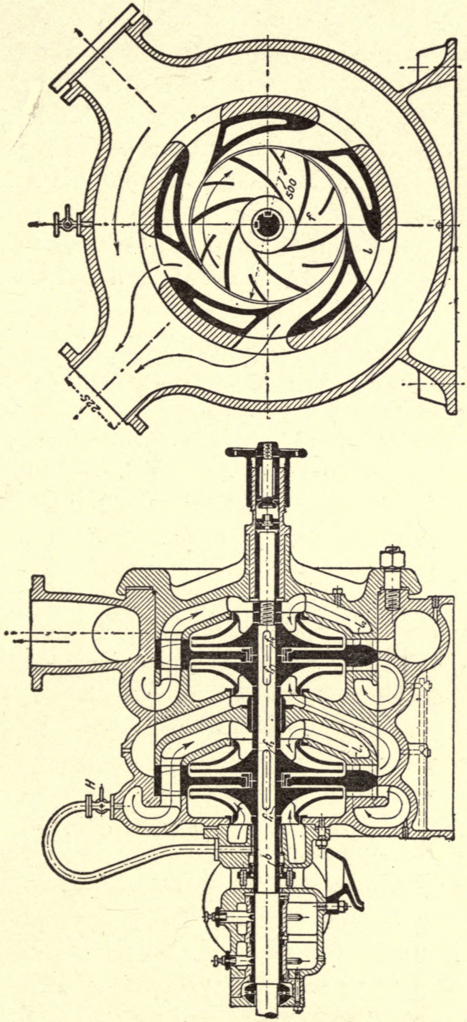
[1] An example was the turbine of the Chase Manufacturing Co., Orange, Mass., shown and tested at the Philadelphia Exhibition, 1876.

recently built and tested in Switzerland, each having four fan-wheels in one case, arranged in series; that is, the first fan-wheel



Figs. 6 and 7.—Quadruple Turbine-pump. Built by Sulzer Bros., Winterthur, Switzerland.

discharges into the suction of the second one, and so on, the water successively passing through all wheels. These fan-wheels have a diameter of 20 ins., and with 890 revolutions deliver 1010 U. S.

gallons of water per minute against a head of 428 ft., which is equal to 107 ft. of head for a pump with single fan-wheel, and give an efficiency of 76%, certainly a great advance in the construction of centrifugal pumps.[1]

As has been said, the turbine-pump is only of recent date, and it is safe to predict that its efficiency will be brought up to that of the turbine, viz., 80 to 82%, at least for heads up to 100 ft.

It is also to be expected that by the use of regulating gates, such as are employed for turbines to change the openings of the guide-buckets, the turbine-pump may be made to discharge at will any quantity of water, from its full capacity to a small fraction thereof, without change of speed or a serious reduction in efficiency.[2]

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[1] Zeitsch. d. V. deutsch. Ing., Nov. 2, 1901, p. 1549; also an abstract in Engineering News, Jan. 23, 1902, p. 66. See also Zeitsch. d. V. deutsch. Ing., Oct. 12, 1901, p. 1448.

[2] Turbine-pumps are discussed in Mr. Elmo G. Harris's paper, "Theory of Centrifugal Pumps and Fans: Analysis of Their Action, with Suggestions for Designers," read before the Am. Soc. C. E., Sept. 16, 1903.

CHAPTER II.

TURBINE PRACTICE IN AMERICA.

Turbine Development in America.—In the United States the development of the turbine has been entirely different from that in Europe. In 1834 Mr. Fourneyron, a French engineer, had brought out the radial outward-flow turbine known under his name, and in 1840 Mr. U. A. Boyden, of Massachusetts, commenced to study and to improve upon this type. Mr. Fourneyron's diffuser was also introduced in America by Mr. Boyden, and is therefore usually known as Boyden diffuser. It should here be stated that the now obsolete diffuser was the forerunner of the conical draft-tube of the present day, and the same principles are underlying the action of both.

Mr. Boyden was soon followed in this work by Mr. James B. Francis. In 1849, however, Mr. Francis built a radial inward-flow or vortex turbine for the Booth Cotton Mills, Lowell, Mass.[1] This turbine, which worked under a head of 19 ft., and when tested showed an efficiency of 79.7%, or practically 80%, may be regarded as the prototype of all American turbines.

The number of revolutions varies as the square root of the heads employed and for the same head the revolutions of different turbines are inversely proportional to their diameters. As all the early turbines were used with low heads, about 20 ft. or less, and as already then, as now, the tendency was to increase the speed of shafting and machinery, builders naturally reduced the turbine diameter.

Mr. Francis's turbine was of the plain inward-flow type, with

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[1] Wood. Turbines, p. 89.

sufficient room in its interior for the water to turn and escape axially. With the continued reduction of the turbine diameter, this interior space became more and more reduced, so that it soon became necessary to turn the water into an axial direction while still in the runner-bucket, or, in other words, to curve the bucket from a radial to a more or less axial direction. This has been going on gradually, as can be seen by comparing the early forms of the Humphrey and the Swain turbines with the present form of the Hercules, New American, Victor, Leffel, and other turbines, which have scarcely more interior space than is required to pass the shaft through and have a much greater part of the runner-buckets in the axial- or parallel-flow direction than in the radial-flow direction, while the runner-buckets have assumed such an intricate shape that it is very difficult to analyze the action of the water while flowing through these buckets, or to mathematically predetermine their shape for given conditions.

Another consequence of the reduction of the diameter is that the inner ends of the buckets, which closely approach the center of the turbine, are located on a very small circle, which limits their number and gives them a very close spacing, while the spacing on the outer circumference becomes so large, 6 ins. and even 12 ins. being not uncommon, that the buckets are unable to properly guide the water as the best efficiency would demand. The area through which the water enters the runner, being the outer circumference of the runner multiplied by the axial dimension of the bucket entrance, decreases, of course, with the diameter of the runner, and with it and in the same ratio decreases the quantity of water passed through and the power developed by the turbine.

To prevent this decrease in entrance area and power, builders have gradually increased the axial dimension of the bucket entrance. Thus the efforts made towards greater speed and power have transformed the plain inward-flow turbine-runner of fifty years ago into the shape now generally employed.

The guide-buckets have not altered so much as the runner-buckets, and of the great variety of gate arrangements that have been tried only the following three have come into general use:

1. The cylinder gate, moving in an axial direction, as used

by Mr. Francis with his early turbines, is now by far the most extensively employed gate.

2. The register gate, a rotating cylinder, having slots which correspond with the outlet openings of the guide-buckets. This gate is not so much used now as some years ago.

3. The wicket gate, having the guide-buckets formed by swinging or otherwise movable blades or vanes. This gate is made in different forms and is next to the cylinder gate, the most widely used arrangement.

Present Turbine Practice in America. The Turbine as a Hydraulic Motor.—For low heads, say up to about 40 ft., the American type of turbine has the great advantage over all other turbine types in common use, that it gives the greatest number of revolutions for a given head and power developed, or the greatest power for a given head and diameter of runner, while the American system of manufacturing only one line of turbines from stock patterns has the great advantage of enabling the builders to fill orders cheaply and quickly.

As already stated, the American turbine has been developed solely by experiment, and the testing-flume at Lowell, and later on that at Holyoke, Mass., may be called the cradle of the American turbine. However, the greatest head available at both these flumes was but 18 ft., and the turbines have therefore been adapted and perfected for such low heads only, while the builders are almost wholly in the dark in regard to the action of their turbines when used with medium or high heads.

This absence of experience seems to have been the principal reason why some of the earlier developments utilize the available head in several stages; for example, a large power in one of the Eastern States, where a head of 80 ft. is utilized in two stages of about 40 ft. each.

So far as the writer is aware, results of reliable tests of American turbines, working under heads much greater than 40 ft., have never been published; and where such tests have been made, the results have been carefully kept from the public. It appears, nevertheless, that a number of similar American turbines of large power and working at Niagara Falls, N. Y., under a head of over 200 ft. are giving a maximum efficiency of some 68%,

while the turbines designed in Europe and working at the Niagara Falls Power Company's plant are said to give 80% efficiency.[1]

At another well-known power-plant, utilizing a head of about 200 ft., the first installation of American turbines gave an efficiency of a little over 40%; the second installation, also of American turbines, but from another maker, gave an efficiency of a little over 60%; while the third installation, which consisted of turbines built in America, but according to European designs, gave an efficiency of nearly 80%.

The low efficiency of the American turbine, when used under high heads, is only what is to be expected, as runner and guide-buckets, constructed to give an efficiency of over 80% under a head of less than 18 ft., cannot possibly also give a high efficiency under high heads, such as 200 ft. or even 100 ft. At the same time, high heads in most cases have only a limited flow of water, thus demanding a higher efficiency for the turbines employed for their utilization than low heads, where the flow is, as a rule, more abundant. In general it must be said that the high power and speed of the American turbines and the custom of filling all orders from stock patterns is also against their use in connection with high heads, as may be shown by two examples, as follows:

While making the specifications for some double turbines, each pair to develop 5500 H.P. under 130 ft. head, and at a speed of 250 revolutions per minute, the writer, in speaking to a well-known turbine builder of the Central States in regard to the matter, found the builder quite willing to bid for and to supply these turbines from his stock patterns. Now, this builder's catalogue shows that a pair of turbines to develop 5500 H.P. under 130 ft. head would make 413 revolutions instead of 250, as required, or that a pair of turbines working under 130 ft. head and running at 250 revolutions per minute would develop 17,312 H.P. The plant was subsequently equipped with turbines of the European type.

A case showing still more forcibly how little the American

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[1] A. van Muyden. Les turbines Fäsch & Piccard à Niagara Falls. Expériences de reception. Bull. de la Soc. Vaudoise des Ing. et Arch., 1895, No. 8.

turbine is adapted to high heads is that of the turbines recently installed by the Cataract Power Co., of Hamilton, Ont., and shown in Figs. 30 and 31. These are single turbines, designed and built in Europe, developing each 3000 H.P. under 256 ft. head, and at a speed of 286 revolutions per minute, and give an efficiency of 80%. No doubt many American turbine builders would have been willing to supply the turbines for this plant from their stock patterns, using, of course, a pair of turbines, as otherwise, under this high head, the end thrust would be too great to be properly taken care of. Using the figures given in a catalogue of McCormick turbines, calculations show that a pair of turbines, to develop 3000 H.P. under 256 ft. head, would have a speed of 1276 revolutions per minute, instead of 286, as required, or that a pair of turbines working under 256 ft. head and running at 286 revolutions would develop 91,565 H.P., or over 30 times the desired amount, and requiring a case 17 ft. in diameter and over 26 ft. long, while the total floor space occupied by the European turbines, up to center of dynamo coupling, is only $11 \times 14\frac{1}{2}$ ft.

Evidently the American type of turbines is not adapted to such high heads, yet the head is not high enough to use impulse turbines to advantage, as may be seen from the following figures, taking the speed at the bucket circle as $v=0.45\sqrt{2gH}$, which is the usual practice. To develop 3000 H.P. under 256 ft. head and at a speed of 286 revolutions would require seven impulse turbines, each having 3 ft. $10\frac{1}{2}$ ins. diameter of bucket circle and three 3-in. nozzles, or 21 nozzles in all. This would certainly be a very complicated arrangement, especially in regard to the regulation of so many nozzles.

It may here be stated that the catalogs of builders of impulse turbines usually give the over-all diameter, that is, from outside of buckets, instead of the diameter of bucket circle, which latter is by far the more important.

When supplying American turbines for high heads, most builders reduce the power of the turbines by using smaller turbines than required for the desired number of revolutions and then draw the speed down to the proper number by means of the governor and the gates. If the power of a turbine is very much in excess of the required amount, turbine builders often

also reduce the axial dimension of the runner-bucket entrance, which, of course, affects the efficiency of the turbine in nearly the same way as would an equal reduction in gate-opening, since only the entrance area is changed, while the rest of the runner-bucket remains as before.

The use of the American turbine in connection with high heads or where special conditions have to be met can thus only be regarded as a makeshift, and turbines of the European type will be preferred in all cases where time and money required for their installation are available, as is shown by the plants of the Niagara Falls Power Co. and the East Jersey Water Co., in the United States, and the Cataract Power Co., already referred to, the Shawinigan Water & Power Co., and the plant at Montmorency Falls, in Canada. It is a matter for wonder that American turbine manufacturerrs will not supply turbines of the European type, where the conditions make their use advisable; but the writer found it practically impossible to induce builders to bid on anything else except the American standard type, even when they were furnished with complete drawings and relieved of all responsibility in regard to the performance of the design. Of course, the writer is aware that some manufacturers make a special high-pressure turbine, but except those built by one firm these machines need not to be considered here, as the guides and runners are the same as those of the regular American turbine, the only difference being in the general arrangement.

But apart from the fact that the American type of turbine is not suitable for high heads, the shape of the runner as now generally used has some features which violate the rules to be observed for obtaining a high efficiency. In Fig. 8 is shown a drawing of a vane and in Fig. 9 a section through one half of the runner and guide of an American type of turbine as built in France,[1] by reference to which the following will be more readily understood.

The water enters the runner-buckets in a radial inward direction, is sharply turned into an axial direction, and leaves the

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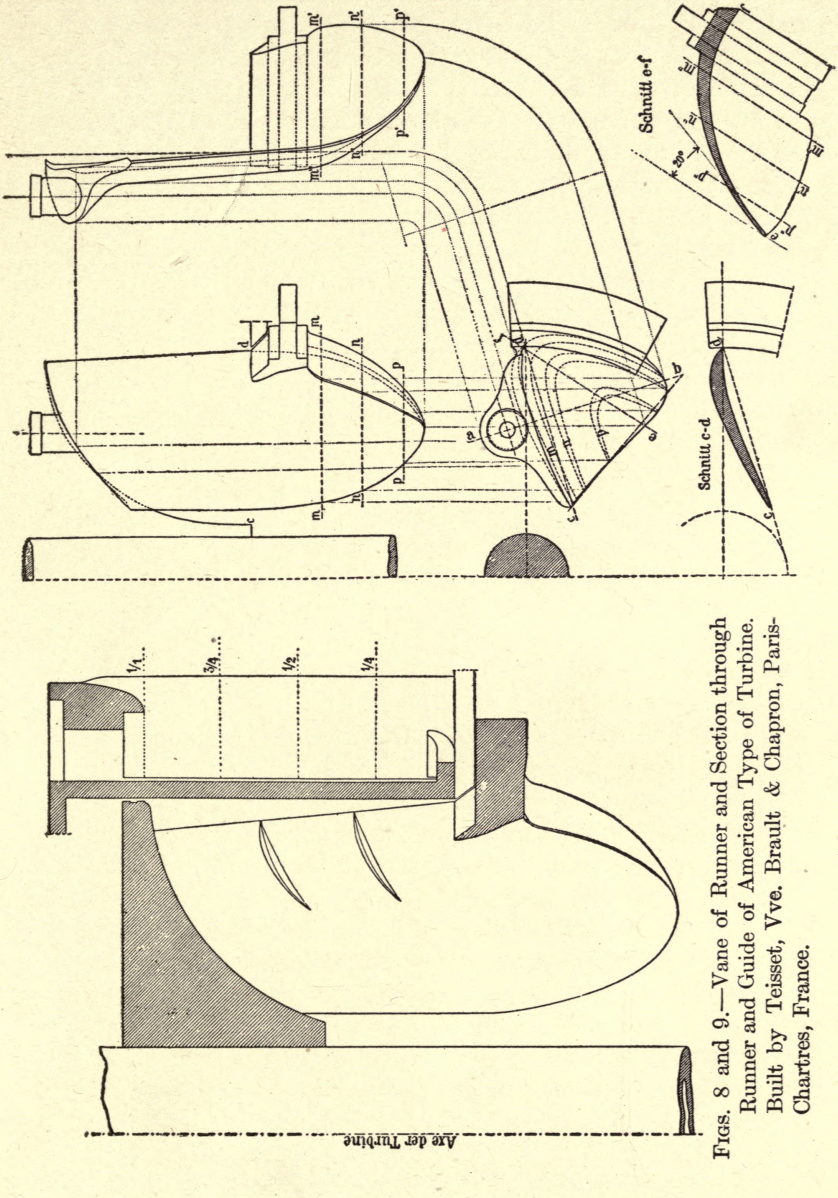
[1] The writer was unable to get satisfactory material for these figures from American builders.



FIGS. 8 and 9.—Vane of Runner and Section through Runner and Guide of American Type of Turbine. Built by Teisset, Vve. Brault & Chapron, Paris-Chartres, France.

buckets, after another turn, in a direction slanting between axial and radial outward, the latter direction, being partly due to the centrifugal force. It will at once be obvious that the two changes in direction cause a loss of head, also that the great length of the buckets means a great frictional resistance or another loss of head. As has already been pointed out, the buckets very closely approach the shaft, thus necessitating a close spacing of the vanes at their inner and a very wide spacing at their outer end, too wide a spacing, in fact, to properly guide the water. However, the cause for the greatest loss in a vortex turbine is that the terminal edge of the vane, being approximately radial, has a speed at its inner end widely different from that at its outer end, but for best efficiency the speed of the terminal edge should be the same as the relative speed of the water at exit from the runner-buckets multiplied by the cosine of the angle of the relative direction of the water or $v_t = c_t \cdot$ cosine γ. This relation of speeds can therefore exist only at one point along the terminal edge of the vane; at all other points there will be a loss of head, increasing with the distance from the point of correct speed. This loss is in the form of velocity-head inside of and up to the point of correct speed, and from that point outward the loss is due to the fact that the runner-vane is moving out of the way of the water before having completely accomplished its object in turning or deflecting the water into the proper direction for discharge. Owing to the fact that the water leaves the runner-buckets in a direction slanting between axial and radial outward, the draft-tube or the draft-elbow or tee has to be much larger in diameter at the point of the runner discharge than the runner itself, as otherwise the water would strike the walls of the tube and be forced to turn too quickly. However, the remedy of increasing the diameter introduces another evil. According to the catalogs of turbine builders, the draft-tube at the point of the runner discharge has an area about twice as large, more or less, as the area corresponding to the diameter of the runner; in other words, the water issuing from the runner has its speed abruptly reduced to one half, and this sudden conversion and shock of course causes loss of head.

American manufacturers have increased the speed and power

of their turbines to the utmost practical limit, and it is safe to predict that any further increase of speed and any further crowding of power into a turbine of given diameter will result in a decided decrease in efficiency. French builders of the American type of turbines, unlike the American builders, which use the same pattern for all heads, have different patterns to suit different heads, as has been stated already, and they also have not increased the speed and power of their turbines to the extent to which American builders have done. The following figures, which are, for the American make, taken from the latest catalog of one of the best-known turbine manufacturers, and, for the French make, taken from material obtained at the Paris Exhibition in 1900, will show this difference.[1]

Diameter of runner at initial or entrance rim:

$$\text{American practice: } D = 1.57 \text{ to } 1.63\sqrt{\frac{Q}{\sqrt{2gH}}}.$$

$$\text{French practice: } D = 1.9 \text{ to } 2.1\sqrt{\frac{Q}{\sqrt{2gH}}}.$$

Ratio of axial dimension of runner-bucket entrance W to diameter of runner at initial or entrance rim:

American practice: $W:D = 1:2$ to $1:2.12$.
French practice: $W:D = 1:2.26$ to $1:2.34$.

Velocity of runner at initial or entrance rim:

American practice: $v = 0.705$ to $0.77\sqrt{2gH}$.
French practice: $v = 0.68$ to $0.75\sqrt{2gH}$.

The majority of the American turbines are now regulated by cylinder gates, as they are not only simple but also cheap, and the latter appears to be the main consideration. It has already been pointed out that cylinder gates do not give good efficiencies with part loads, except when additional crowns are used. This,

[1] Scheiz. Bauz., Feb. 9, 1901, p. 53; also Zeitsch. d. V. deutsch. Ing., Dec. 28, 1901, p. 1842.

however, is not possible with the American type of runner, owing to the complicated shape of the buckets. A number of manufacturers therefore cast on to the face of the runner-vanes, near the entrance edge, projections which in shape somewhat resemble a half of a lens. These projections are shown in Fig. 9, and are intended to prevent the water from turning too abruptly, but are also expected to act, to a limited extent, in the same way as additional crowns.

With full gate-opening, the American type of turbine is a vortex turbine, working with a great amount of reaction, but this changes as the gate is closing, and with very small gate-openings the water flows radially inward and issues from the runner-buckets along the edge near and parallel to the shaft and the turbine works without any reaction, but with action only. As the reaction is decreased, the initial angle of the runner-bucket, β, should, for best efficiency, also decrease, and one American turbine-builder has therefore divided the axial lengths of the vane at the bucket entrance into three parts, each having a different initial angle. These angles do not change gradually, but by abrupt steps, which, however, steadily diminish in width and disappear entirely some distance in from the entrance. That part of the bucket entrance which is the last to be covered by the closing of the gate has, of course, the smallest initial angle, as it is here where the reaction ceases and action only is at work. The steps or shelves formed by the changes of angles have to a small degree the effect of the lens-shaped projections mentioned above.

It seems that the hydraulic efficiency of most American vortex turbines would be increased by running them at lower speed than they are listed in the catalogs. This was clearly shown by some experiments made recently by a German turbine-builder to ascertain the merits of different types of inflow reaction turbines when working under low heads.[1] One of the turbines tested was a 30-in. vortex turbine built by a well-known American manufacturer, and which, under a head of 7 ft., should run, according to the catalog, at 143 revolutions per minute and give

[1] Zeitsch. d. V. deutsch. Ing., June 13, 1903, p. 841.

an efficiency of 80% with full gate. The actual efficiency at 143 revolutions was 64% with full gate- and 63% with 0.75 gate-opening. This turbine gave the best efficiency when running at 124 revolutions, viz., 63.5% with full gate- and 72.5% with 0.75 gate-opening; thus by reducing the speed by 13.3% the maximum efficiency was increased by 8.5%.

From theoretical considerations it also appears that the efficiency of the American vortex turbine, both at full and part gate-opening, would be increased by leaving off the scoop-shaped discharge end of the runner-vanes, giving this end a form similar to the one shown in Fig. 25 or 26, and having a different angle of relative discharge, denoted by γ in Fig. 10, at different distances from the centre of the runner, so that the equation cosine $\gamma=\frac{v_t}{c_t}$ will be at least approximately satisfied at every point along the discharge edge of the runner-vanes. This would not only allow the shape of the runner-buckets to be more readily analyzed, but it would also decrease the length of the path of the water in the runner-buckets and with it decrease the friction loss. It would also permit to have the entrance area of the draft-tube of the same size as the discharge area of the runner and the water issuing from the runner-buckets would have both the direction and speed consistent with the requirements for best efficiency and agreeing with the theory of turbines.

German builders of turbines have greatly improved the efficiency of the American vortex turbine by properly designing the runner-buckets and giving them a shape as indicated in Fig. 22.

Present Turbine Practice in America. The Turbine as a Machine.—So far the American turbine has only been considered as a hydraulic motor, but it may be well to consider it also as a machine.

Turbines have not yet come to be regarded as important machines, and builders therefore employ in their construction such materials and workmanship as are only used for second- and third-class machinery, in sharp contrast to European practice, where turbines are constructed with the same care as high-class steam-engines.

For installations with vertical turbines, the builders, besides

furnishing the turbine proper, can only prepare plans and give advice in regard to the general arrangement, while the choice of material and workmanship rests with the purchaser of the turbine. One sees, therefore, a great variety of settings, from the crazy wooden flume stuck to the outside of a dilapidated mill to the modern concrete and steel construction.

The use of turbines on horizontal shaft, made possible by the use of the draft-tube, has been rapidly increasing during recent years, and while this arrangement means a very great advance in general, practically no improvements have been made in the constructive details of horizontal turbines.

The case for such turbines consists usually of cast-iron heads and a shell made of low quality steel plate, barely thick enough to withstand the interior water-pressure and having two light steel I beams fastened along its bottom, supposed to stiffen the whole. The turbines themselves and such main bearings and gate shaft bearings as are located inside of this case are held in place and fastened to or braced against the thin shell plate by cast-iron or gas-pipe supports. The regulating-gate rigging and other parts are often bolted or riveted to the outside of the shell, while the outside main bearings are either carried on little shelves cast onto the heads of the case or are bolted to the draft-tube elbow or elbows, or supported by yokes resting on the above-mentioned I beams.

It is only reasonable to assume, especially when it is considered that the rivet- and bolt-holes in the plates forming the case are often badly matched, and the drift-pin is freely used during erection, that the water-pressure in the case due to the head under which the turbine is working will somewhat alter the shape of the case from what it is while the case is empty, in which state, of course, all assembling, riveting, bolting, and adjusting was done, and this alteration of shape must naturally throw everything that is fastened to the case out of alinement, besides setting up stresses not provided for in the design.

The incessant surging or oscillating of the water in the penstock and in the draft-tube, due principally to changes in the gate-opening in regulating the turbine, causes the water-pressure in the case to alternately rise above and sink below the normal

pressure due to the working-head. These variations are greatest where long penstocks and draft-tubes are used, and they keep the case and everything fastened to it continually working or moving and under constantly varying stresses, a state of affairs certainly not productive of easy running or long life of the turbine.

It may be mentioned here that the surging and with it the rise and fall of the pressure in the turbine-case may even take place with a steady load on the turbine and a constant gate-opening. In such cases this surging will usually be found to be due to wave motion and eddies at the penstock entrance if the penstock is short, or to eddies in the penstock itself if the latter is long and not properly designed.

When obliged to employ steel-plate cases, the writer has mitigated this evil somewhat by using thicker plates and deeper I beams than were proposed by the builders.

Another fault of steel-plate cases is the difficulty of properly shaping them, and in consequence the water while passing through such cases is subjected to abrupt changes in direction and speed, and to reduce the loss in working-head caused by such changes the cases are made larger than would be required if easy waterways could be obtained.

A great number of plants have been installed with two or more turbines in one case and admitting the water through the end of the case. If local conditions make it necessary that the water shall flow towards the turbine in an axial direction, the open turbine-chamber would be much the better arrangement; but where circumstances, such as high heads, cost, etc., demand the employment of a case, the latter should be considerably larger in diameter than the diameter of the turbines would otherwise require, so as to have sufficient area for the axial flow of the water to reach the second and any further turbines that may be in the case without unnecessary loss of head.

Below the turbines the axial flow of the water is completely blocked by the draft tube or tubes, and this part of the cross-sectional area of the case is therefore lost. By setting the turbine shaft below the centre line of the case, this lost area is reduced and the useful area above the turbines increased, so that sufficient area for the axial flow of the water may be obtained, with

little or no increase in diameter over that required for a case with side inlet.

To have thrust and other bearings, gate-shafts, gears, racks, guide-rollers, set-screws, and other moving or adjustable parts inside the turbine-case where they cannot be inspected or adjusted, as is now the universal custom with builders, must also be considered faulty construction, the more so as it could be easily avoided.

The size of the turbine-shaft as generally supplied by builders is barely sufficient to transmit the power generated by the turbine, whereas the shaft should be considerably stronger than the power alone would demand, particularly for turbines on horizontal shafts, and here again a greater excess of strength is required for a pair of turbines on one shaft than for a single turbine.

Weak and springy turbine-shafts and bad workmanship mean large clearances, and these in turn mean leakage and therefore loss of water and reduction in turbine efficiency.

When ordering a large turbine on a horizontal shaft for a head of 40 ft. or more builders will usually advise the purchaser to use a pair of turbines on one shaft to avoid the end thrust, while for heads of 80 ft. or more most builders will decline to assume any responsibility in regard to the thrust-bearing if a single turbine is used, claiming that no such bearing can be made that will properly take care of the end thrust of large single turbines under high heads. This is of course a preposterous assertion in the view of the daily experience of thousands of ocean steamers, where the whole propelling power is converted into end thrust and 15,000 H.P. and more are applied to a single shaft. It must be said, however, that thrust-bearings to work satisfactorily must be well designed, of good material and liberal proportions, and of superior workmanship.

However, the American type of turbine when considered as a machine also has its advantages, the foremost of which are that the turbine is simple, has but few parts, and is easily taken apart and assembled.

Causes of Lack of Progress among American Turbine-builders.—The backward state of the art and the scarcity of improvements in American turbine construction is certainly surprising when compared with the advanced state to which nearly

every other line of machinery has been brought in America. There is this to be said for the turbine-builder, however—that he would have very little sale for a high-class turbine. The fault rests chiefly with the purchaser, who generally believes that water-power costs little or nothing, and that any turbine which will turn his machinery is good enough.

It is also very much to be regretted that the practice of testing turbines in the flume at Holyoke has become so universal, as although these tests may give very accurate figures, showing how the efficiencies of different sizes and makes of turbines compare with each other when tested with a head of less than 18 ft., and under the most favorable conditions, they do not give any information whatever about the real or absolute efficiency the turbine will show when properly set up and connected in its intended place and working under a head perhaps five or ten times greater than that available at Holyoke.

The Holyoke tests, as usually made, with the turbine set vertical in an open turbine-chamber, will show an efficiency closely approaching the hydraulic efficiency of the turbine, but the same turbine that has shown at Holyoke an efficiency of, say, 82% under a 17-ft. head, when installed in its intended place, mounted on a horizontal shaft and enclosed in a case, may easily give an efficiency ten or more per cent less than shown at Holyoke, although the head used might be the same in both cases. This difference may be due to increased friction in the bearings, faulty design of the case and draft-tube, poor workmanship, bad alinement, etc., and it is little consolation to the owner that the turbine has given good results at Holyoke.

To test turbines in their place and under working conditions, as is nearly always done in Europe, would in most cases cost but little more than the Holyoke test, but the results would be immensely more valuable to the purchaser, to the builder of the turbines, and to the engineering profession generally, provided, of course, the tests are made by competent persons.

Two cases may be mentioned to show the slight value of the Holyoke test. A year ago the writer made a contract for some 1400-H.P. turbines to work under a head of 110 ft., and a well-known builder offered to guarantee 80% efficiency. Upon

being informed that the tests were to be made with the turbines in place, he modified his guarantee to 80% at Holyoke and 73% in place. The other case is that of a large power plant in Canada, containing when fully equipped 72 turbines of 200 H.P. each, working under a head of 11 ft. The builder guaranteed the customary 80% efficiency, shipped one of the turbines to Holyoke, and was able to show test results of about 83%. The engineers in charge of the construction of the plant in question took at random a turbine from the lot received and had it tested at Holyoke, but the efficiency was only 72%, and it required a great deal of polishing and improving to bring the efficiency of this turbine up to 80%, and the conclusion is naturally that only the two turbines tested are up to the guaranteed 80%, and all the rest have an efficiency of about 72%.

The results of efficiency tests made at Holyoke cannot be directly compared with the results obtained by tests made in Europe, as has been shown repeatedly by having turbines tested at Holyoke and retested in Europe, the latter test invariably giving a much lower efficiency than the Holyoke tests.

It is not the writer's intention to discuss here the intricate questions involved in the testing of turbines, nor to say which of the test results are nearest to the actual efficiency; but it may be stated here that European engineers maintain this difference to be principally due to the different methods of water measurement.

As an example may be mentioned a 16-in. turbine bought by a European builder from one of the largest and best-known turbine manufacturers in the United States, the object being to take up the construction of American turbines if the high efficiencies claimed could be verified. The turbine was duly tested at Holyoke in the spring of 1900 and retested in Germany in the spring of 1901. The tests in Germany were made by one of the highest authorities on turbines,[1] and the testing flume was equipped with the latest and most refined instruments.

The efficiencies of the American turbine shown at the Holyoke

[1] Professor A. Pfarr, of the Polytechnicum in Darmstadt, formerly for many years chief engineer of the works of J. M. Voith, turbine manufacturer, Heidenheim, Germany.

test and at the German test are given below, and for comparison the efficiencies now commonly realized with European reaction turbines, when tested in the same manner as the American turbine has been in Germany, are also given.[1] These figures will bear out the writer's statement that the shape of the American type of runner is not adapted to give the best efficiency, especially with part gate-opening.

Discharge	1.0	0.9	0.8	0.7	0.6
American turbine, Holyoke test	0.81	0.795	0.765	0.725	0.67
American turbine, German test	0.718	0.703	0.693	0.658	0.591
European turbine, German test	0.80	0.81	0.82	0.81	0.80

Discharge	0.5	0.4	0.3	0.2
American turbine, Holyoke test				
American turbine, German test	0.491	0.358	0.121	
European turbine, German test	0.79	0.76	0.70	0.60

The discharge is here the proportional amount actually discharged and should not be confounded with gate-opening. The smallest discharge tested at Holyoke was 0.6. The total weight on the turbine step was 25% greater during the Holyoke tests than during the German tests.

The turbine tests made at the Centennial Exhibition in Philadelphia, 1876, have never been given much credit by either American or European engineers.

As the term gate-opening is at present employed indiscriminately for both gate-opening and discharge, and as many users of turbines when comparing efficiencies think that with, say, one half gate-opening the turbine will only pass one half of the amount of water used with full gate, the writer would here call attention to the great difference between the two. For example, the American turbine referred to above discharged 0.6 of the amount of water used with full gate, with a gate-opening of only 0.321; that is, the discharge was nearly twice as great as the gate-opening.

[1] Zeitsch. d. V. deutsch. Ing., June 7, 1902, p. 845; also a defence of the Holyoke test by Mr. Clemens Herschel and reply to same in Zeitsch. d. V. deutsch. Ing., Nov. 22, 1902, p. 1788.

The importance of a high efficiency in cases where water is bought or power sold may be easily shown. Supposing the water required to develop one gross or theoretical horse-power costs \$10.00 per year, which is about the average of prices paid in the New England States, then the net or effective horse-power costs \$14.30 with 70% and \$12.50 with 80% efficiency of turbines, or a saving of \$1.80 per effective horse-power in favor of the turbine with the higher efficiency. With 5% interest on capital invested and 10% for depreciation, taxes, etc., or a total of 15%, which is very high, the above \$1.80 would pay the interest on \$12.00, or, in other words, the turbine giving 80% efficiency could cost \$12.00 more per horse-power, or a 1000-H.P. turbine could cost \$12,000 more, than a turbine giving only 70% efficiency, without increasing the cost per effective horse-power per year. As a 1000-H.P. horizontal double turbine of the American type, for 80 or 100 ft. head, complete with case and draft-tube and erected in place, costs about \$6000, and will give about 70% efficiency, a properly designed and built turbine giving 80% could cost \$18,000 without being more expensive than the other; but as such a turbine could be bought for about \$10,000, the higher-priced turbine really means a saving in capital invested to the amount of \$8000, or \$8 per horse-power.

Or supposing that power is sold at \$15 per year for one effective mechanical horse-power at the turbine-shaft and that the water-supply is limited, the amount of water required to develop one effective horse-power with 70% efficiency will give 1.143 H.P., worth \$17.14 per year, with 80% efficiency. The difference of \$2.14 in favor of the turbine with the higher efficiency is equal to an interest of 15% as above, on \$14.27, or a 1000-H.P. turbine giving 80% efficiency could cost \$14,267 more than a turbine giving 70% without being more expensive.

Other causes that have retarded the progress in turbine construction may be mentioned, such as the belief of purchasers that the turbine is a very simple machine, and that they know all about it, and therefore do not need the service of a hydraulic engineer; that the few hints given in turbine catalogs are all that is necessary to know, and that any third-class machinist or millwright who has once helped to install a turbine is a com-

petent hydraulic engineer. That the ignorance of the purchasers in regard to turbines is well known to builders is shown by the unreasonable claims and statements made in turbine catalogs.

But hydraulic-power engineers, where consulted in connection with water-power developments, are also often to blame for not properly explaining to their clients the advantages of a high-class turbine, and the turbine specifications prepared by engineers are as a rule poor, superficial, and insufficient, and no effort is made to ascertain whether the specifications have been complied with or not. It may be said that there are many so-called hydraulic engineers who do not know any more about turbines than what they can learn from builders' catalogs, and who apparently have not the least idea that anything better can be produced in the line of turbines than is generally furnished by the builders.

Another reason that turbines have advanced so little in America, and that their design has not yet been placed upon a scientific basis, is the lack of attention paid to the subject by the engineering schools and the want of an up-to-date text-book on turbine design. In all European colleges the theory and the design of turbines is taught with the same care as the theory and the design of steam-engines, while in many American colleges, it seems, the theory of turbines is only casually mentioned in the lectures on the flow of water, while turbine design is not taught at all.

The hydraulic-power engineer, when called upon to develop a large water-power, say under 200 ft. head, finds himself in a very awkward position. If he chooses single turbines, there will be trouble with the thrust-bearings.

If the turbines are to drive dynamos or other direct-connected machinery, and are selected to give the right speed, their power will usually be enormously too large, and if selected to give the right power, the speed as a rule will be much too high, and special dynamos or machinery would have to be built to suit the turbines, while the efficiency of the latter would be low in any case. If he goes to Europe for his turbines, there will be delays and much extra expense.

For a number of plants with whose construction the writer was connected the cost of the turbines amounted to between

4 and 10% of the total cost of the development. Now, while the turbine is the most important element in a development, its cost is comparatively small and the increased price of a high-class turbine would be hardly noticeable in the total cost of the development. Why then should the hydraulic-power engineer be obliged to design the whole plant with only one point in view, that is, to suit the standard turbine patterns? And has the time not arrived yet when the greatly varying conditions of head, amount of water, size and speed of turbine-units, etc., demand more than just one line of patterns for the hydraulic engineer to choose from?

The general cry for cheapness and the striving to meet competition by reducing the cost of production seems to have done more harm in the manufacture of turbines than in most other lines. Economy is one of the principles on which an industry must be based to successfully meet competition, but to lower the quality of material and workmanship and to reduce the hydraulic and mechanical efficiency of turbines for mere cheapness' sake is certainly a wrong application of the principles of economy.

Thousands of dollars are often spent in the general construction work for a development, to gain a few additional feet of head, where a greater gain in power might be obtained by expending a couple of hundreds of dollars more on the turbine equipment.

CHAPTER III.

CLASSIFICATION OF TURBINES.

Before going into the details of the turbine types best adapted to the different heads, it may be well to give here the general characteristics and properties and the advantages and disadvantages of the different classes of turbines. Turbines are classified, according to the manner in which the water performs its work in flowing through them, into reaction and action turbines. The action turbines are again divided into two sub-classes, viz., action turbines with free deviation and action turbines with limited buckets or limit turbines.

The principal difference between reaction and action turbines is in the velocity with which the water issues from the guide-buckets, or, what is the same, in the presence or absence of pressure at the clearance. The principles of working of each of the different classes and sub-classes of turbines may be embodied in a variety of types, but only the types that are of importance in modern turbine practice will here be dealt with.

It may here be well to recall some of the laws of hydraulics or the mechanics of a fluid:

Water stored under a static head or pressure contains a certain amount of energy in the potential form. To make this energy available for the development of power the static head or pressure of the water is converted into velocity in the turbine; that is, the potential energy is converted into kinetic energy and it is the duty of the turbine to absorb this kinetic energy either by reaction or action or by both, and to convert the energy thus absorbed into power and to discharge the water as an inert mass or nearly so, leaving only enough velocity in the water to carry it away from the turbine. It will therefore be seen that the energy

contained in the water may exist either in the potential or kinetic form or in both forms and part or the whole of the energy may be converted from one form into the other; also that the head may exist either in the form of static head, which is pressure, or in the form of velocity or in both forms, and that pressure can be converted into velocity and velocity into pressure.

It should be noted here that the head used in the formula for the discharge velocity of water, $\sqrt{2gH}$, is always the head available or effective at the turbine and that the velocity found by this formula is the theoretical velocity, which has to be multiplied by the coefficient of discharge for the bucket, nozzle, or opening to obtain the actual velocity. For example, for a properly constructed guide-bucket the coefficient is from 0.95 to 0.97, or the speed of the water is from 0.95 to $0.97\sqrt{2gH}$.

If c is the velocity of the water corresponding to the available head and c_a the absolute velocity with which the water leaves the runner-buckets, then of all the energy contained in the water the portion absorbed by the turbine is $\frac{c^2-c_a{}^2}{c^2}$ or $\frac{H-h_a}{H}$; therefore c_a should be as low as practicable.

The speed of the runner or the speed factor always refers to the initial or entrance rim of the runner.

Within reasonable limits the closer the vanes are spaced—that is, the greater the number of buckets in the guide and runner—the better will the water be guided, but with an increase in the number of buckets the frictional losses in guide and runner are also increased. In practice it is usual to have the number of runner-buckets different from the number of guide-buckets.

The Reaction Turbine.—Suppose a narrow vessel rectangular in plan is mounted at the end of a horizontal arm, which latter is fastened to a vertical shaft free to revolve, the longer dimension of the vessel being tangential to the circle described by the vessel when turning with the shaft. Suppose also the vessel to be filled with water, then there is a hydrostatic pressure against the sides, the pressure against opposite sides balancing each other. If an opening is provided in one of the smaller sides and at the same time the water level in the vessel is kept constant by a con-

tinuous supply of water, then the velocity of the jet of water issuing from the opening will be $c_t=\sqrt{2gH}$, in which H is the head of water in feet, above the centre of the opening. With the water flowing out of the opening at the rate of Q cu. ft. per second, the weight of water discharged is 62.3 Q lbs. per second and the mass of this is $\frac{62.3Q}{g}$. To give to this mass an acceleration of c_t ft. per second requires a force or pressure in lbs. of $P=\frac{62.3Q}{g}c_t$.

This force P, required to give the acceleration c_t to the water jet, exerts an equal and unbalanced pressure or reaction upon the side opposite to the opening, applied at a place corresponding to the centre of the opening and tending to move the vessel in a direction opposite to that of the water jet. With the vessel moving in the direction opposite to that of the jet of water and with the speed of v_t ft. per second, the pressure or reaction equals

$$P=\frac{62.3Q}{g}(c_t-v_t),$$

and the horse-power developed equals

$$\text{H.P.}=\frac{P\cdot v_t}{550}.$$

If the speed of the vessel is equal to that of the water jet, or $v_t=c_t$, then the theoretical hydraulic efficiency would be 100%; or, in other words, with the vessel or bucket moving at the same speed as the water jet, but in opposite direction, the water has no horizontal motion when compared with a stationary point; that is, the absolute horizontal velocity would be $c_a=c_t-v_t=0$, and the water would simply be left behind and drop vertically, as if the bottom were pulled out from under it.

Thus for a hydraulic motor, as here described, working with reaction only, the speed of the vessel for the best or theoretically perfect hydraulic efficiency would be equal to the speed of the water jet, or $v_t=c_t=\sqrt{2gH}$, as in this case all the kinetic energy is taken out of the water and the latter discharged quite

motionless or dead. The speed factor for such an ideal reaction motor, having a velocity of $v_t = \sqrt{2gH}$, would therefore be 1.

However, from the formula for the pressure or reaction P, it will be seen that with the speed of the vessel or bucket v_t equal to the speed of the water jet c_t, the pressure P becomes zero, that is, such an ideal reaction motor can do no useful work.[1]

Segner's and Barker's turbines[2] were hydraulic motors working with reaction only in the manner here outlined.

The reaction turbine is also sometimes called pressure turbine.

The types in which reaction turbines may be built are the axial flow, radial outward flow, radial inward flow, and the vortex or American type. Only the inflow and vortex types are considered here.

The characteristic of the reaction turbine is that the water issues from the guide-buckets with a speed smaller than that due to the head, or $c < \sqrt{2gH}$; that is to say, at the moment the water reaches the clearance, only part of its head is in the form of velocity and the remainder in the form of pressure. In the runner-buckets this remaining pressure is also converted into speed, so that the water when issuing from the runner-buckets has attained the velocity due to the whole head. It may be well here to consider the effect of water flowing through the runner-buckets of a reaction turbine. For this purpose let us assume the runner as stationary, neglecting here the fact that the usual bucket angles would not permit the water to flow properly into and through the runner-buckets if the runner is stationary. Let us also suppose that one half of the head of the water is converted into velocity in the guide-bukcets, as is the common practice, then the speed of the water at the exit from the guide-buckets or the clearance would be $c = \sqrt{2g \times 0.5H} = 0.707\sqrt{2gH}$, and the pressure would be that due to the remaining half of the head. In the runner-buckets the water is deflected from the discharge direction of the guide-buckets to the discharge direction of the runner-buckets or through an angle of $\varepsilon = 180° - (\alpha + \gamma)$. (See Fig. 10.) The water thus works by action while flowing through the runner-buckets.

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[1] Wood. Turbines, pp. 50 to 55. [2] Wood. Turbines, p. 48.

The speed c of the water at the clearance is increased in the runner-buckets by the remaining half of the head and at the exit from the runner-buckets this speed will be $c_t=\sqrt{(2g\times 0.5H)+c^2}$, or as $c^2=2g\times 0.5H$, we have

$$c_t=\sqrt{2(2g\times 0.5H)}=1.4142\sqrt{2g\times 0.5H}$$
$$=1.4142\times 0.707\sqrt{2gH}=\sqrt{2gH}.$$

The equation $c_t=\sqrt{(2g\times 0.5H)+c^2}$ is derived from the fundamental formula for accelerated motion, $f=\frac{w}{g}\times\frac{v_2-v_1}{t}$, as follows, the factor 62.3 being the weight of water per cubic foot:

$$0.5H\times 62.3=\frac{62.3}{g}\times\frac{c_t+c}{2}\times c_t-c;$$

$$0.5H=\frac{c_t^2-c^2}{2g};\quad c_t^2=(2g\times 0.5H)+c^2;$$

therefore $$c_t=\sqrt{(2g\times 0.5H)+c^2}.$$

From this it will be seen that while the half of the head which is the first to be converted into speed gives the water a velocity of $c=0.707\sqrt{2gH}$, the remaining half of the head only adds to this velocity the amount of $c_t-c=0.293\sqrt{2gH}$, or as $c:c_t-c=0.707:0.293=1:0.4142$, the speed added by the remaining half of the head is only 0.4142 of the speed due to the half of the head which is the first to be converted into velocity, or, in other words, the discharge velocity of water varies with the square root of the head.

This is due to the fact that the volume of water accelerated per second for each square foot of discharge opening of the guide-buckets is c cu. ft., while the volume accelerated for each square foot of discharge opening of the runner-buckets is $c+c_t$ cu. ft., thus giving the same proportion as the accelerations, but inversed,

or $$c:c+c_t=c_t-c:c=0.293:0.707=0.4142:1.$$

Therefore $$0.707\sqrt{2gH}\times c=0.293\sqrt{2gH}\times c+c_t;$$

that is to say, the amount of work done per second by each half of the head is the same.

The water thus works by reaction while leaving the runner-buckets, the reaction being due to the increase in the velocity of the water while flowing through the runner-buckets.

With the runner moving at its proper speed, the actual velocity with which the water enters the runner-buckets will not be c, but $c_e = c \cdot \text{sine}\, \alpha$, if the entrance angle of the runner-buckets β is 90° and the velocity with which the water leaves the runner-buckets will be correspondingly decreased.

The proportion of the work done by action and reaction is indicated by the speed factor, as the higher the speed factor of a turbine the greater is the reaction, while a turbine works with action only if the speed factor is less than 0.5. The proportion of work done by reaction and action depends entirely on the shape of the guide- and runner-buckets and the bucket shape in turn depends on the angles α, β, and γ. (See Fig. 10.) The angles

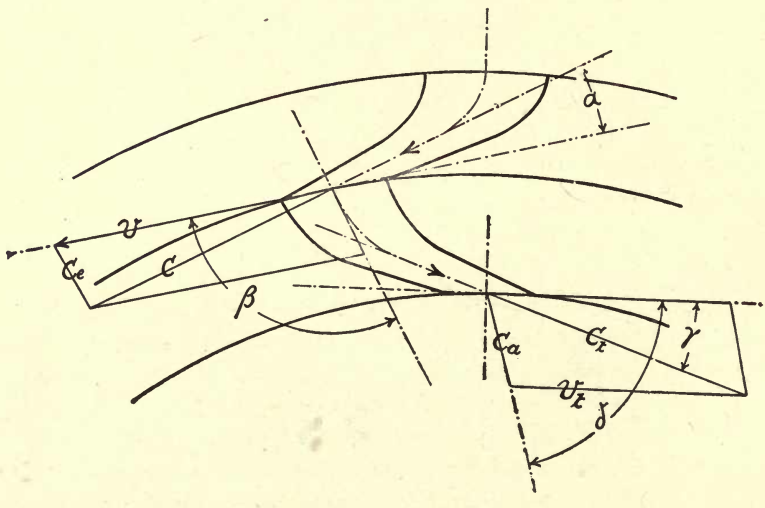

FIG. 10.—Diagram of Guide- and Runner-bucket, showing angles and velocities.

α, β, and γ of reaction turbines are interdependent, and if one is changed, the other two have also to be altered, or the efficiency

and proper working of the turbine will be affected. For axial-flow turbines the following equation has to be observed to obtain the best efficiency: cotang α = cotang β + cotang γ.[1] With radial-flow turbines, the diameter of the initial or entrance rim D and the diameter of the terminal or exit rim D_t of the runner have to be taken into consideration. The width between crowns of radial turbines usually increases from the initial rim to the terminal rim, and this has also to be considered. If W is the width or clear space between crowns at the initial rim of the runner and W_t at the terminal rim, then we have for radial inward-flow turbines:[2]

$$\text{cotang}\,\gamma = \left(\frac{D_t}{D}\right)^2 \frac{W_t}{W} (\text{cotang}\,\alpha - \text{cotang}\,\beta)$$

or

$$\text{cotang}\,\alpha = \text{cotang}\,\beta + \text{cotang}\,\gamma \left(\frac{D}{D_t}\right)^2 \cdot \frac{W}{W_t}.$$

Also, for a turbine to work with reaction, angle α must be less than one half of angle β,[3] or

$$2\alpha < \beta.$$

A turbine has ceased to work with reaction and has become an action turbine if

$$2\alpha = \beta.$$

Again, the more the angle β exceeds 2α the greater will be the amount of reaction with which the turbine is working.

Theoretically reaction turbines require no guide-buckets, as shown by Segner's and Barker's turbines above mentioned, but they are now always provided with guide-buckets.

The size of the terminal angle of guide-buckets α does not affect the efficiency, but is usually made as small as practical considerations will permit; that is to say, the direction of the water

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[1] Taschenb. Huette., vol. 1, p. 735.

[2] Taschenb. Huette., vol. 1, p. 740.

[3] Mueller. Francis-Turbinen., p. 70.

at exit from guide-buckets is as nearly a tangent to the runner as practicable, and it is therefore the size of the initial angle of the runner-buckets β, which determines the amount of reaction and the speed of the turbine.

The terminal angle of the runner-buckets γ should be made, for best efficiency, as small as practicable; that is to say, the relative direction of the water issuing from the runner-buckets should be as nearly as possible a tangent.

However, with the decrease in the size of the angles α and γ, the clear-passage areas at the terminal ends of the guide- and runner-buckets decrease also, and therefore for turbines of equal diameter and width and shape of crowns, smaller angles α and γ mean smaller water capacity or power of the turbine.

European practice for radial inward-flow turbines is to make both the angles α and β from 15° to 24°, but circumstances will often require these angles to be as much as 30° or even larger.

The angle of the actual or absolute direction of the water at exit from the runner-buckets δ should be made, for best efficiency, equal to 90° or $v_t = c_t \cdot \text{cosine}\, \gamma$. If this angle is made less than 90°, there will be an increased loss of velocity-head in the discharged water, and if made larger than 90°, there will be a loss due to the runner-vanes moving out of the way of the water before having completely accomplished their object in turning or deflecting the water into the proper direction for discharge.

The greater the amount of reaction with which a turbine is working, the greater is the speed of the runner, as may readily be seen from Fig. 10. The amount of reaction with which a turbine is working does not affect its efficiency.

Therefore to obtain the highest speed with a given head and runner diameter a reaction turbine with a large angle β would be chosen, and to get the lowest speed with a given head and runner diameter an action turbine would be chosen.

European practice for standard patterns of radial inflow turbines is to make the angle β equal to 90° and to have the velocity and the pressure of the water, at exit from guide-buckets or at the clearance, each that due to one half of the head, or speed of water $c = \sqrt{2g \times 0.5H}$, which is approximately $c = 0.71\sqrt{2gH}$, and the corresponding speed of the runner at the entrance rim is

$v=c \cdot \text{cosine}\,\alpha$, or approximately $v=0.67\sqrt{2gH}$, which would require α to be $19\frac{1}{3}°$ [1].

The head of the water remaining as pressure at exit of guide-buckets or at the clearance is therefore

$$h_r=H-\frac{c^2}{2g}.$$

The loss of head in the guide-buckets, due to friction and the conversion of pressure into velocity, will vary, according to conditions, between 6 and 10% of the head converted into velocity, and the actual speed of the water at exit from guide-buckets therefore will be $c=0.95$ to $0.97\sqrt{2gh}$, in which h is that part of the head which is converted into velocity, or for reaction turbines, converting one half of the head into velocity, $c=0.95$ to $0.97\sqrt{2g\times0.5H}=0.675$ to $0.69\sqrt{2gH}$, and $v=0.64$ to $0.65\sqrt{2gH}$.

However, the amount of reaction and with it the speed factor varies considerably in general practice and the speed factor may be anywhere above that of limit turbines, or about 0.50 up to 0.90 or even more, but usually is between 0.57 and 0.75.

Below are given the speed factors for some types of inflow turbines:

American or vortex turbines: 0.70 to 0.77.

American or vortex turbines of French manufacture: 0.68 to 0.75.

European type, standard practice: 0.67.

As extreme cases may be mentioned the 2500-H.P. double turbine shown in Fig. 25, with a speed factor of 0.82, and the 1000-H.P. triple turbine seen in the cross-section of power-house, Fig. 19, with a speed factor of 0.925, the latter being perhaps the highest ever used in modern turbine practice.

The speed of the runner at the terminal rim is $v_t=\frac{vD_t}{D}$, and the corresponding speed of the water at exit from runner-buckets and relative to these buckets is $c_t=\frac{v_t}{\text{cosine}\,\gamma}$, if δ is equal to 90°.

[1] Mueller. Francis-Turbinen, p. 78.

The velocity-head corresponding to the absolute velocity with which the water leaves the runner-buckets is lost, except such portion of it as may be recovered by means of a draft-tube; therefore for best efficiency this absolute speed c_a should be as low as practicable without interfering with the proper working of the turbine; or, in other words, of the energy contained in the water before entering the turbine, as much as possible should be absorbed by the runner before the water is allowed to escape.

With angle δ equal to 90°, the absolute exit velocity is $c_a = v_t \cdot \text{tang}\, \gamma$, or $c_a = c_t \cdot \text{sine}\, \gamma$; but this velocity increases with both a decrease or an increase in the size of the angle δ.

With turbines working in the air, the absolute exit velocity c_a may be very low, or the water might even simply drop out of the runner-buckets by gravity, but for turbines working under water or in a case, some speed has to be retained in the discharge to free the runner, displace the surrounding water, overcome the friction and carry off the discharged water, as otherwise the turbine would be working against a back pressure.[1]

The water flowing through the runner-buckets of a reaction turbine completely fills these buckets, at least with the turbine working at full or nearly full capacity.

The guide- and runner-buckets must be formed by easy curves and all changes in direction and speed of the water must be made gradually, and the water issuing from the guide-buckets must meet the vanes of the runner-buckets without shock or impact. To avoid such impact the following equations have to be observed for speeds and bucket angles:

$$c = \frac{v \cdot \text{sine}\, \beta}{\text{sine}\, (\beta - \alpha)};$$

$$v = \frac{c \cdot \text{sine}\, (\beta - \alpha)}{\text{sine}\, \beta};$$

$$ce = \frac{v\ \text{sine}\, \alpha}{\text{sine}\, (\beta - \alpha)}.$$

[1] Mueller. Francis-Turbinen., pp. 76 and 86.

The losses of energy in reaction turbines are between 17 and 27% and are due to:[1]

Hydraulic resistances	10 to 14%
Velocity of water discharged from turbine	3 " 7%
Leakage at clearance	2 " 3%
Shaft friction	2 " 3%
Total losses	17 to 27%

The remaining energy or efficiency would therefore be between 83 and 73%.

The effect of the speed-regulating gates on the water flowing through a radial inflow reaction turbine may here be considered. Let us assume the turbine to be running with the gate or gates half closed.

1. The cylinder gate will cause the jets of water issuing from the guide-buckets to be reduced to one half of their width or axial dimension, while the thickness of the jets or their circumferential dimension remains the same for all gate-openings. But in the runner-buckets the jets will spread in width and decrease in thickness, so that the water jets do not fill either the width or the circumferential dimension of the runner-buckets.

2. The register-gate will cause the jets of water issuing from the guide-buckets to be reduced to one half of their thickness, so that the water jets while filling the width do not fill the circumferential dimension of the runner-buckets.

3. The wicket gate will cause the water jets issuing from the guide-buckets to be reduced to one half of their thickness and also to be diverted from their direction of flow at full gate, denoted by α in Fig. 10, to a direction tangential to a circle, which may be of a smaller, equal, or larger diameter than the entrance rim of the runner, according to the arrangement of the wickets. If the circle to which the water jets are tangential is of a larger diameter than the entrance rim of the runner, the jets will evidently not strike the runner at all, but will flow around with the runner and at a speed exceeding the speed of the latter, and it is this excess of speed and the pressure of the water issuing from

[1] Mueller. Francis-Turbinen., p. 18.

the guide-buckets which force the water into the runner-buckets. With a comparatively small entrance angle of the runner-buckets β, the water jets will not fill the circumferential dimension of these buckets, while with a comparatively large entrance angle β, the water jets will fill the runner-buckets, but will flow through them with a decreased velocity, the decrease in velocity being in direct proportion to the decrease in the volume of water admitted by the regulating-gates.

From the above considerations the following conclusions may be drawn:

A reaction turbine working with part gate and having the runner-buckets not completely filled with water will work with action only, the reaction effect being lost, and there will be no pressure at the clearance, except the slight amount required to overcome the friction and the centrifugal force of the water, and thus cause it to enter the runner-buckets.

A reaction turbine working with part gate and with action only will give a lower efficiency with such part gate than a turbine designed to work with action only at all gate-openings; that is, an action turbine.

As there is practically no pressure at the clearance of a reaction turbine when working with part gate and with action only, the water will issue from the guide-buckets with a speed almost as great as the speed due to the whole head or $c=\sqrt{2gH}$, and therefore the percentage of the volume of water passed by such part gate-openings is greater than the percentage of the gate-opening itself. For example, a reaction turbine in which the water issues from the guide-buckets with a velocity of $c=\sqrt{2g\times 0.5H}$ at full gate would pass at full gate $0.71\sqrt{2gH}$ cu. ft. of water per square foot of discharge opening of the guide-buckets. The same turbine, when running with half gate-opening would pass $0.5\sqrt{2gH}$ cu. ft. of water per square foot of discharge opening of the guide-buckets, or the water passed at full and half gate would be as $1:0.71$.

The effect of the water completely filling the runner-buckets of a reaction turbine, but flowing through these buckets at a reduced speed, when running with part gate, will be referred to in connection with the effect of throttling-gates.

The principal advantages of the reaction turbine are:

The turbine gives a good hydraulic efficiency.

The hydraulic efficiency is the same with the turbine working in the air or under water or in a case. It should here be noted, however, that the mechanical efficiency of the turbine will be from $1\frac{1}{2}$ to $2\frac{1}{2}\%$ less when working under water or in a case than when working in the air, owing to the friction of the runner in the water.

Part of the head may be utilized by means of a draft-tube without affecting the efficiency.

With a given head and runner diameter, the reaction turbine may be designed to give up to twice the speed or number of revolutions obtainable with the action turbine, which is of advantage in connection with low heads.

With a given head and runner diameter, the speed or number of revolutions of the reaction turbine may be varied between wide limits without affecting the efficiency by varying the bucket angles.

The principal disadvantages of the reaction turbine are:

The regulating-gates, to give good efficiency with part gate-opening, have to be much more complicated than those of action turbines, and even then the efficiency with small gate-openings is less than for action turbines.

The turbine does not give a good hydraulic efficiency if used as a partial turbine, and can therefore not well be utilized for high heads.

The water while passing the clearance is under pressure, which means leakage of water and thus reduction in efficiency.

The Action Turbine.—It will be obvious to every one that a jet of water striking a stationary or relatively stationary surface and being deflected by the latter will exert a pressure against that surface, the pressure being the greater the greater the angle is through which the jet is deflected. If P is the pressure in pounds exerted by a water jet against a surface, such as a bucket of an impulse turbine, and ε is the angle through which the jet is deflected, then $P=\frac{62.3Q}{g}(c-v)(1-\text{cosine } \varepsilon)$, in which v is the speed of the buckets at the bucket circle and 62.3 the weight of a cubic

foot of water. The minimum and maximum values for P would be with ε equal to zero or no deflection and ε equal to 180°—that is, reverse of direction—or

$$P=\frac{62.3Q}{g}(c-v)(1-\text{cosine } 0°)=0$$

and

$$P=\frac{62.3Q}{g}(c-v)(1-\text{cosine } 180°)=\frac{2\times 62\ 3Q}{g}(c-v)=$$
$$=2\times 1.9372Q(c-v).^1$$

The horsepower of the impulse turbine would therefore be equal to $\text{H.P.}=\frac{Pv}{550}$.

For best efficiency the theoretical speed of a hydraulic motor working with action only would be equal to one half the speed of the water or $0.5\sqrt{2gH}$; that is to say, the speed factor would be 0.5. This will at once be seen from the following considerations: If the speed of the bucket circle of an impulse turbine is one half the speed of the water or $v=0.5c$, then the water strikes the buckets with a velocity relative to the buckets of $c_e=c-v=0.5c$, and being deflected by the buckets through 180°—that is, reversed —leaves the buckets with the same relative speed, or $c_t=c_e=0.5c$, and travels in the opposite direction, or backward, at that speed. It will therefore be obvious that the water leaving the buckets and traveling backwards with a realtive speed of $c_t=0.5c$, the buckets at the same time traveling forward with a speed of $v=0.5c$, will have an absolute speed of $c_t-v=0$, or, in other words, the water has been brought to rest, compared with some stationary point, and as all the energy in the water had been in the form of kinetic energy, all the energy has been absorbed by the runner.

Action turbines are also often called pure action turbines or action turbines with free deviation, to distinguish them from the action turbines with limited buckets. The types in which such turbines may be built are the axial-flow, radial outward-flow, and radial inward-flow types, which may all be either full

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[1] Taschenb. Huette., vol. 1, p. 262.

or partial turbines, also the impulse and the spoon types, the latter resembling in its features both the radial outward-flow and the impulse types. Only the outflow and impulse types are considered here.

The characteristic of the action turbine is that the water issues from the guide-buckets or nozzle with the full velocity due to the head, or $c=\sqrt{2gH}$; that is to say, at the moment the water leaves the guide-buckets or nozzle all the pressure or potential energy of the water has been converted into kinetic energy and the turbine therefore works with action only.

Much that has been said about the reaction turbine applies also to the action turbine.

For outflow action turbines the angles α and β are usually made $2\alpha=\beta$, but β may also be smaller than 2α. The angles α and γ should be as small as practicable—that is, each should be as nearly a tangent as possible—for the reason that the smaller α and β, the greater is the angle of total deflection of the water and therefore the greater is the efficiency, the angle of total deflection being $\varepsilon=180-(\alpha+\gamma)$.

European practice is to make α about 12° and γ about 13° for high heads and small volumes of water, increasing to α about 30° and γ about 28° for low heads and large volumes of water.

The angle of the actual or absolute direction of the water, at exit from the runner-buckets δ, should be made, for best efficiency, equal to 90° or $v_t=c_t \cdot \text{cosine}\, \gamma$. If this angle is made less than 90°, there will be an increased loss of velocity-head in the discharged water, and if made larger than 90°, there will be a loss due to the runner-vanes moving out of the way of the water, before having completely accomplished their object in turning or deflecting the water into the proper direction for discharge.

Impulse turbines may be regarded as action turbines; for example, an impulse turbine deflecting the water sidewise, that is in a more or less axial direction like the Pelton, may be regarded as a partial action turbine having on the same shaft two runners of the axial-flow type, placed in close contact with their inlet or entrance sides, discharging axially in opposite directions and having a larger bucket spacing than used for the common axial-flow type. There are also no crowns, the radial spreading of the

water-jet being prevented by raised edges on the buckets, while each nozzle takes the place of a guide-bucket.

For properly designed impulse turbines the angle α should be only a few degrees, thus giving a high efficiency, while the angle γ measured in the plane in which the water-jet is deflected varies according to the shape, size, and spacing of the buckets and the diameter of the bucket circle, as the angle γ has to be large enough that the jet leaving a bucket will clear the bucket following.

The diameter of the bucket circle should not be less than twelve times the diameter of the nozzle discharge-opening for a round nozzle, or the diameter of a circle equal in area to the discharge-opening of a rectangular nozzle.

The loss of head in the guide-buckets or nozzle, due to friction and the conversion of pressure into velocity, will vary, according to conditions, between 6 and 8% of the head, and the speed of the water at exit from guide-buckets or nozzle therefore will be $c=0.96$ to $0.97\sqrt{2gH}$. In straight smooth nozzles of impulse turbines the loss of head will not be greater than 4 to 6% and the speed of the water therefore $c=0.97$ to $0.98\sqrt{2gH}$.

The actual speed of the runner of turbines of European design is for outflow action turbines $v=0.43$ to $0.47\sqrt{2gH}$, and for impulse turbines the actual speed at the bucket circle is $v=0.44$ to $0.47\sqrt{2gH}$.

The best efficiency, the absolute speed c_a, with which the water leaves the runner-buckets should be as low as practicable, without interfering with the proper working of the turbine; or, in other words, of the energy contained in the water before entering the turbine, as much as possible should be absorbed by the runner before the water is allowed to escape.

With the angle δ equal to 90°, the absolute exit velocity is $c_a=v_t\cdot\text{tang}\,\gamma$ or $c_a=c_t\cdot\text{sine}\,\gamma$, but this velocity increases with both a decrease or an increase in the size of the angle δ.

The water flowing through a runner-bucket of an action turbine with free deviation does not completely fill this bucket, but shoots along the face of the vane, leaving an empty space between the water and the back of the next vane. A ventilation has to be provided for this empty space, either through the inlet- or

entrance-opening of the bucket or through holes properly located in the crowns of the runner, as otherwise this empty space would fill with dead water, which would seriously interfere with the proper working of the turbine and thus reduce the efficiency.

As for good efficiency all action turbines have to work in the air, the absolute exit velocity c_a may be very low, or the water might almost simply drop out of the runner-buckets by gravity.

The guide- and runner-buckets must be formed by easy curves and all changes in direction made gradually. All changes in speed in the guide-buckets or nozzle must be made gradually. The water issuing from the guide-buckets of an action turbine must meet the vanes of the runner-buckets without shock or impact, and to avoid such impact the following equations have to be observed for speeds and bucket angles:

$$v=\frac{c\cdot\text{sine }(\beta-\alpha)}{\text{sine }\beta}; \quad c_e=\frac{v\text{ sine }\alpha}{\text{sine }(\beta-\alpha)}.$$

The principal advantages of the action turbine are:

The turbine gives a good hydraulic efficiency. The regulating-gates are very simple and the turbine gives a good hydraulic efficiency even with small gate-openings.

The turbine gives almost as high an efficiency, if built as a partial turbine, as if built as a full turbine.

With a given head and runner diameter the action turbine gives the lowest speed or least number of revolutions of any class of turbine, which is of advantage in connection with high heads.

With a given head and volume of water the number of revolutions of the action turbine may be varied between wide limits by varying the runner diameter and building the turbine as a full or partial turbine, as the runner diameter and volume of water may demand.

The water, while passing the clearance, is not under pressure, therefore no water is wasted by leakage at the clearance.

The principal disadvantages of the action turbine are:

The turbine must always work in the air, as when working under water the runner-buckets cannot be ventilated and consequently the efficiency is much reduced.

If a draft-tube is to be used in connection with an action turbine, an air-admission valve has to be provided to supply the air for ventilating the runner-buckets and to regulate the water level in the turbine-case or draft-tube, so as to be always below and clear of the lowest point of the bucket-ring of the runner.

The Limit Turbine.—The limit turbine is usually considered as an action turbine and therefore also called action turbine with limited buckets, but as a matter of fact the limit turbine stands in the middle or forms the dividing line between the action and reaction turbine.

The types in which limit turbines may be built are the axial-flow, radial outward-flow, and radial inward-flow types, but the inflow turbine appears to be the most promising and therefore the type that will be most generally used in the future.

The limit turbine, combining as it does many of the advantages of both the action and the reaction turbine, has as yet not received the attention which it deserves, and it is to be hoped that this turbine will soon find a more extended application.

The characteristic of the limit turbine is that, even if designed to work with action only, the water flowing through the runner-buckets completely fills these buckets, at least with the turbine working at full or nearly full capacity.[1]

Most of what has been said about the action and reaction turbines applies also to the limit turbine.

The angles α and β for limit turbines working with action only are made $2\alpha=\beta$, but β may also be made larger, up to about $2\alpha+10°=\beta$, thus giving the turbine a slight amount of reaction. Angles α and γ are made as small as practicable and δ is made 90° or slightly more or less, the same as stated for action turbines. The loss of head in the guide-buckets is from 6 to 8%, as given for action turbines and the actual speed of the runner of limit turbines is $v=0.48$ to $0.49\sqrt{2gH}$.

The absolute discharge velocity c_a may be very low for turbines working in air, but for turbines working in water, c_a must be sufficient to free the turbine from the discharged water.

The guide- and runner-buckets must be formed by easy curves,

[1] Meissner Hydraulische Motoren, vol. 2, p. 321.

changes in direction and speed of the water must be made gradually, and the general curvature and the clear areas at different points of the runner-buckets must be such as not to force the water-jet from its natural shape. To avoid the impact of the water against the vanes of the runner-buckets, the following equations have to be observed:

$$v=\frac{c\cdot\text{sine }(\beta-\alpha)}{\text{sine }\beta}; \quad c_e=\frac{v\cdot\text{sine }\alpha}{\text{sine }(\beta-\alpha)}.$$

The principal advantages of the limit turbine are: The turbine gives a good hydraulic efficiency. The hydraulic efficiency is the same with the turbine working in the air or under water or in a case, although the mechanical efficiency will be from $1\frac{1}{2}$ to $2\frac{1}{2}\%$ less when working under water or in a case than when working in the air, owing to the friction of the runner in the water.

Part of the head may be utilized by means of a draft-tube without affecting the efficiency.

Very simple regulating-gates may be used, and the turbine gives a good hydraulic efficiency even with small gate-openings.

The turbine gives nearly as high an efficiency, if built as a partial turbine, as if built as a full turbine, but a partial limit turbine must always work in the air and if working in a case, with draft-tube, an air-admission valve has to be provided to keep the water level in the case or draft-tube below the lowest point of the runner, as otherwise the efficiency would be reduced. With a given head and runner diameter the limit turbine gives a speed or number of revolutions almost as low as the action turbine, which is of advantage in connection with high heads. With a given head and volume of water, the number of revolutions of the limit turbine may be varied between wide limits by varying the runner diameter and building the turbine as a full or partial turbine as the runner diameter and volume of water may demand.

The water while passing the clearance is under little or no pressure, therefore little or no water is wasted by leakage at the clearance.

The following table gives a comparison of the power developed

by different designs of inflow turbines, all working under the same head of 1 meter (3.28 ft.) and running at 74 revolutions per minute.[1] The original metric measure has been retained to avoid odd fractions. The vortex turbine is of European design; that is, a modification of the American type of turbines and the high- and medium-speed inflow turbines are reaction turbines like the vortex. The dimensions given are, of course, for the entrance rim of the runner.

Design.	Diameter of Runner, in Millimeters.	Width between Crowns, in Millimeters.	Horse-power.	Efficiency.
Vortex turbine	1000	550	19.1	75%
High-speed inflow turbine	700	150	3.88	82%
Medium-speed inflow turbine	700	100	1.87	80%
Limit turbine	600	50	0.58	75%

This table shows that the high-speed inflow reaction turbine gives the best efficiency, the efficiency decreasing for designs having a higher or lower speed or giving a greater or smaller power; also, for turbines working under the same head the table shows that:

With a given number of revolutions the vortex turbine gives the greatest and the limit turbine the least power.

With a given power the vortex turbine gives the highest and the limit turbine the lowest number of revolutions.

[1] Zeitsch d. V. deutsch. Ing., June 13, 1903, p. 846.

CHAPTER IV.

STEAM-TURBINES.

As is well known to engineers, the first reaction steam-turbine was invented about 120 B.C. by Heron, a Greek living at Alexandria, Egypt, and the first action steam-turbine of the impulse type was invented by Giovanni Branca, an Italian, in the year 1629. Both these devices may be regarded merely as toys, and it was only during the last two decades of the nineteenth century that steam-turbines were developed into commercial machines. However, this development has been a rapid one, as both the physical properties of the steam and the laws governing the action of a fluid in a turbine were well known. Thus in less than twenty years the steam-turbine has been advanced from a toy to a machine which, running condensing, gives a steam efficiency superior to a high-grade compound condensing Corliss engine of the same power.

It is not the writer's intention here to enter into the thermodynamic conversion of energy, but simply to consider the mechanical action of steam in a turbine.[1]

In general it may be stated that the same laws and principles governing hydraulic action and reaction turbines apply also to turbines working with any other fluid, either liquid or gaseous; but due consideration must be given to the differences in physical

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[1] For an exhaustive study of the steam-turbine, the excellent work by Dr. A. Stodola, "Steam Turbines," New York, 1905, is strongly to be recommended. This book not only treats fully on the theory and practice of the steam-turbine and the thermodynamic principles involved, but also describes and illustrates the different types of turbines manufactured, and gives direction for designing and calculating the dimensions of the various types of steam-turbines.

properties of liquid and gaseous fluids. Taking water and steam as examples of liquid and gaseous fluids, although saturated steam is not a true gas, the following three principal differences in physical properties will be found.

1. Water acting in a turbine has, for all practical purposes, always the same weight per cubic foot; that is, water is not expansive, while steam is expansive, changing its volume and density or weight per cubic foot with every change in pressure and temperature.

If p is the pressure in pounds per square inch above the vacuum or the absolute pressure, t the temperature Fahrenheit, and 0.622 the specific density of gaseous steam, that of air being one, then we have:

Weight in pounds per cubic foot or density of steam:

Saturated steam $L=\dfrac{p^{0.941}}{330.36}$.

Gaseous, that is highly superheated, steam $L=\dfrac{2.7074p\times 0.622}{459.2+t}$.

Volume in cubic feet per pound of steam:

Saturated steam $V=\dfrac{330.36}{p^{0.941}}$.

Gaseous, that is highly superheated, steam $V=\dfrac{459.2+t}{2.7074p\times 0.622}$.

2. The pressure or head of the water is directly due to its weight, while the pressure of steam is due to its expansive force produced by the quantity of heat contained in the steam, but the pressure per square inch may also be considered as a head. This head is equal to the height in feet of a column of steam having an area of base of 1 sq. in. and a uniform density corresponding to the given absolute initial pressure, the weight of which column is equal to the given difference between initial and exhaust steam pressure per square inch, or head of steam in feet $H=\dfrac{144(p-p_1)}{L}$, in which p_1 is the exhaust or back pressure. In an analogous manner the atmospheric back pressure

against which a hydraulic turbine is discharging may be partly removed or a partial vacuum produced by the weight of a hanging water column in a draft-tube, while the atmospheric back pressure against which a steam-turbine is exhausting may be partly removed by removing part of the quantity of heat contained in the steam by means of condensation.

3. Water under any pressure or head will flow out of a nozzle with a velocity of $c=\sqrt{2gH}$. The velocity of steam flowing from a greater absolute pressure p into an atmosphere of a less absolute pressure p_1 increases as the difference in pressure increases. Following the same general law as the velocity of water, or $c=\sqrt{2gH}$, and substituting for H its equivalent as given above, we have [1] $c=\sqrt{2g\times\frac{144(p-p_1)}{L}}$. However, this law only applies as long as the external pressure p_1 is 58% or more of the absolute internal pressure p, or $p_1 \geqq 0.58p$, and the velocity of the steam shows practically no further increase with the fall of the external pressure below 58% of the internal pressure even to the extent of a perfect vacuum. In flowing through a nozzle of the best form, steam expands to the external pressure so long as it is not less than 58% of the internal pressure. For an external pressure of 58% and for lower percentages, the ratio of expansion in the nozzle is 1 to 1.624.

It will therefore be seen that the velocity of steam flowing out of a nozzle or opening is due both to the pressure and the expansion of the steam.

The following is a general consideration of the steam-turbine and its advantages and disadvantages, especially as compared with reciprocating steam-engines.

Theoretically a jet of steam issuing from a properly designed nozzle will develop as much energy as if the same steam were

[1] The equation here given for c expresses the general law for the outflow of gases, and in particular holds true for the outflow of atmospheric air, but does not give accurate results when applied to steam. Grashof's formulas for the outflow of steam are considered as the most accurate. However, as these formulas are very intricate, they are not given here, but may be found in Grashof's "Theoretische Maschinenlehre," vol. 1, paragraphs 111 to 113; also in Taschenb. Huettc., vol. 1, p. 289.

expanded behind an engine-piston, but in a turbine expansion can be carried much further than it is practicable in an engine. With a reciprocating engine there is no gain in efficiency to be secured by expanding the steam to less than 5 or 6 lbs. absolute pressure on about 20 to 18 ins. of vacuum, as not only would the low-pressure cylinder become very large and thus the friction loss very great, but as each end of the cylinder is alternately connected with the steam-inlet and the exhaust and the exhaust temperature is very low with a high vacuum, the cylinder condensation and re-evaporation will be correspondingly high and the loss greater than the gain due to further expansion. In a steam-turbine, a back pressure as low as 1 or 2 lbs. absolute, or about 28 to 26 ins. of vacuum, can be easily obtained, as there is no increased friction loss due to a high vacuum, nor is there the loss due to condensation and re-evaporation, as the steam in a turbine always flows in the same direction, while on the other hand, according to the Carnot cycle, a low exhaust pressure and temperature means a high thermal efficiency, or

$$\eta=\frac{T_1-T_2}{T_1};$$

in which T_1 is the absolute inlet and T_2 the absolute exhaust temperature.

The steam-turbine also shows a much higher economy with light and overloads than the reciprocating engine, and this is especially the fact when the turbine is run condensing and is then due, in a large measure, to the high vacuum that is used in connection with turbines. Steam-turbines will operate economically with loads of from 50% to 125% of the full load, and in some types even with from 25% to 150% of the full load.

The steam-turbine shows a considerable gain in economy when using superheated steam, and this is almost entirely due, according to Prof. Thurston, to the reduction of the surface or skin friction between the vanes and the steam. As there are no interior rubbing or wearing surfaces, stuffing-boxes or packings, etc., requiring lubrication and exposed to the live steam, superheated steam of much higher temperature may be used than is

advisable with reciprocating engines, and owing to the absence of interior lubrication the condensed steam may be returned directly to the boiler, which is especially of advantage in connection with marine steam-turbines. The absence of interior rubbing surfaces does away with the friction and wear of the principal working parts.

The steam-turbine has few working parts, the motion is rotary and the power directly applied, the torque or turning moment is uniform and the speed is uniform for all parts of the revolution and therefore is smooth-running and free from vibrations, thus requiring little or no foundations.

The steam-turbine is compact, requires little space, and is easily and cheaply installed.

Owing to the high velocity with which steam issues from nozzles or guide-buckets, the speed of steam-turbines is also very high, and while this is an advantage in driving dynamos, centrifugal pumps, blowers, etc., as it permits the use of small sizes of such machines, the high speed is a disadvantage in the majority of cases. But there is another serious obstacle to the high speed, viz., the limited strength of the material of which the runner is made. For example, with 150 lbs. gage pressure and 28 ins. vacuum, steam would flow out of an expanding nozzle with a velocity of nearly 4000 ft. per second, and to utilize this speed in a single-stage action turbine, such as the De Laval, would require a rim speed of about 1800 ft. per second, which is higher than any material could be subjected to without bursting. To reduce the speed of the runner, most designers employ a series of turbines, utilizing a part of the pressure or velocity of the steam in each individual turbine or stage of the series.

With the high speeds employed, it has also been found necessary to permit the shaft and runner to revolve about their axis of gravity instead of their geometrical axis, to allow for the fact that shaft and runner may be out of balance, due to lack of homogeneity of the material or faulty workmanship or both, as otherwise the turbine would quickly jar itself to pieces.

The speed regulation of steam-turbines also presents some difficulties. Most turbines are now regulated by some device, which throttles the steam and thus causes it to have a reduced

pressure in the admission-chamber. Like with hydraulic turbines, it is more difficult to regulate reaction steam-turbines so as to show good efficiencies at part load than action steam-turbines.

A great disadvantage of the steam-turbine, especially for marine work, is that it cannot be reversed, except by having a separate set of guide- and runner-buckets for both the forward and backward motion.

Steam-turbines require the very highest class of material and workmanship, owing both to their high speed and the small size and large number of their parts. A 400-H.P. Westinghouse-Parsons steam-turbine, for example, has 14,978 guide-vanes and 16,095 runner-vanes.

It may be well to mention here that many designs of gas- and oil-turbines have been proposed; that is, turbines in which gas or oil is burned with air, as in the reciprocating gas- and oil-engines. One of the designs was brought out by Mr. De Laval, but he has since abandoned the idea. The principal difficulty is that the air required for combustion has to be compressed by large, slow-speed reciprocating air-compressors with their adherent losses, and theoretical investigations have shown that gas- and oil-turbines cannot be expected to show a greater economy than steam-turbines.[1]

The steam-turbines built in the United States at present, viz., the Westinghouse-Parsons, the De Laval, the Rateau, and the Curtis, are briefly described in the following:

The Westinghouse-Parsons steam-turbine shown in Fig. 11 in longitudinal section is the American modification of the Parsons steam-turbine, which is the only commercially successful reaction steam-turbine.[2]

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[1] V. Lorenc, "Waermeausnutzung der Heissluftturbinen," Zeitsch. d. V. deutsch. Ing., Feb. 24, 1900, p. 252.

[2] For more detailed description of the Westinghouse-Parsons turbine, see Francis Hodgkinson, "Steam Turbines," in proceedings of the Engrs. Soc. of Western Pennsylvania, Nov. 1900; also "Some Theoretical and Practical Considerations in Steam Turbine Work," read before the Am. Soc. M. E., June 2, 1904; an abstract of the latter printed in Engineering News, June 9, 1904, p. 553. For English forms of Parsons stationary and marine turbines see Stodola, "Steam Turbines," pp. 271, 311, and 313.

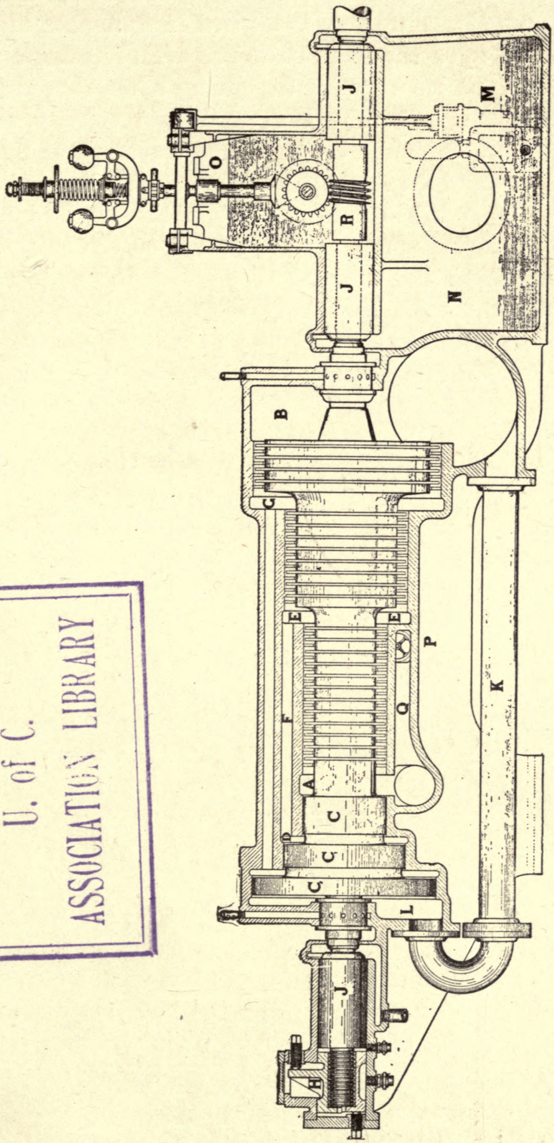

FIG. 11.—Longitudinal Section of Westinghouse-Parsons Steam-turbine. Built by the Westinghouse Machine Co., Pittsburg, Pa.

The Westinghouse-Parsons steam-turbine is a series turbine; that is to say, the steam-pressure is utilized in a number of axial turbines, mounted upon the same shaft and each forming a stage or step in the successive expansions. Being a reaction turbine, the steam-pressure is not fully converted into velocity or the steam expanded in the guide-buckets, but leaves these buckets still under pressure, and the velocity of the steam is further increased and its volume expanded in flowing through the runner-buckets, which absorb the velocity and kinetic energy of the steam, and the steam issues from the runner-buckets with the proper pressure for entering the guide-buckets of the next turbine or stage. As the steam leaves the guide-buckets still under pressure, there will of course be a certain amount of leakage through the clearance between guide- and runner-buckets and around the runner-to the guide-buckets of the next stage.

The steam enters the turbine through the admission-chamber *A* in Fig. 11, and flows to the right through the successive stages of guide- and runner-buckets, the passage areas of these buckets increasing as the volume of the steam increases by expansion. After a number of turbines have been passed, it becomes necessary, in order to further increase the passage areas of the buckets, to increase the diameter of the set of turbines following, and this increase may be repeated several times, as shown at *E* and *G*, the steam finally entering the exhaust-chamber *B*.

As all reaction turbines have a considerable end thrust, special provisions are made to take care of the same. For this purpose revolving balance-pistons *C*, *C*, and *C* are provided, which correspond in diameter to the three sets of turbines, each of which is subjected to the same pressure as exists in the admission-chamber of its corresponding set of turbines, the pistons for the second and following sets being connected with the corresponding admission-chamber by balance ports, as shown at *F* for the second set. These balance-pistons, it may be mentioned here, are not in contact with the surrounding cylinder, and while this will cause a certain loss by leakage, it will also prevent the friction loss, except the loss due to the friction of the water resulting from the condensation of steam. To equalize the exhaust pressure with the pressure against the back of the piston corresponding to the last

set of turbines, the pipe connection K is provided. A thrust-bearing, shown at H, is provided to take care of any unbalanced end thrust and thus prevent end motion.

By means of the by-pass valve P and the port Q, the full steam-pressure in the first admission-chamber A may be admitted to the second admission-chamber E, to provide for an emergency overload of about 60% and to permit a turbine running condensing to develop its full power, even should the condenser be inoperative and the turbine exhausting into the atmosphere. The opening of the by-pass naturally reduces the economy of the turbine.

The heavy turbine shaft with its runners is allowed to revolve about the axis of gravity, by the peculiar arrangement of the bearings and therefore a flexible coupling, shown at R, becomes necessary to transmit the power developed by the turbine.

The lubricating oil drains to the reservoir N and the pump M raises it to the reservoir O, from which it flows to the rubbing surfaces by gravity.

The speed regulation of the Westinghouse-Parsons turbine is interesting, as, like for hydraulic turbines, a relay governor is employed. The steam, except with the maximum load, does not flow continuously into the admission-chamber A, but enters in consecutive puffs, the duration of the puffs being in accordance with the load. The purpose of this arrangement is to always have the steam enter the admission-chamber with the full initial pressure, but it will easily be seen that in the intervals between puffs the pressure in the admission-chamber will at once fall below the initial pressure and that the pressure in the second and remaining admission-chambers will be little different from what it would be if the regulation were accomplished by simply throttling the steam.

A disadvantage of the Westinghouse-Parsons turbine is that owing to the great number of turbines, the small clearances necessary in reaction turbines, and the fact that some of the turbines revolve in a high steam-pressure, there will be a comparatively large friction loss, which will be much increased if the steam is not superheated and thus water of condensation is allowed to accumulate in the turbine.

The De Laval steam-turbine is an axial-flow action turbine with partial feed, in which the whole steam-pressure is converted into velocity in a single set of nozzles and this velocity and kinetic energy absorbed by a single runner.[1]

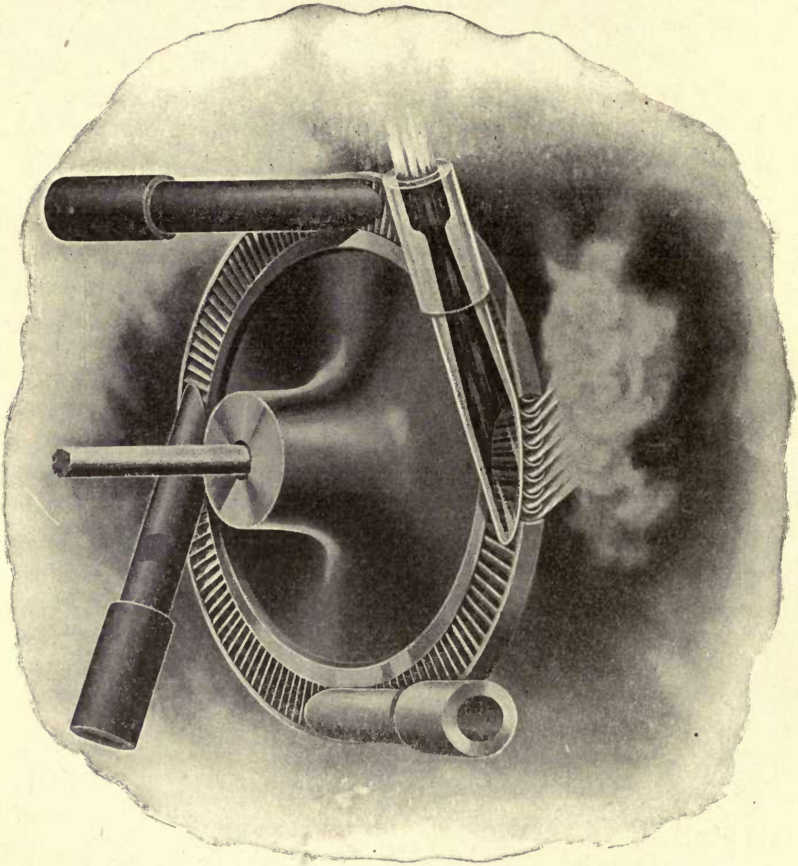

FIG. 12.—Runner and Nozzles of De Laval Steam-turbine. Built by the De Laval Steam Turbine Co., Trenton, N. J.

The pressure of steam while the steam is flowing through an ordinary nozzle will decrease only to 58% of the absolute

[1] For more detailed description see Stodola, "Steam Turbines," p. 216; also Mr. E. S. Lea's paper, "The De Laval Steam Turbine," read before the Am. Soc. M. E., June 2, 1904, and printed in Engineering News, June 9, 1904, p. 551.

initial pressure, and in order that the whole expansion of the steam should take place while flowing through the nozzle, Mr. De Laval devised the expanding or diverging nozzle shown in Fig. 12. The steam in flowing through the throat of this nozzle expands and decreases in pressure to 58% of the absolute initial pressure and then enters and flows through a diverging tube, which may be considered as an infinite number of infinitely short nozzles of successively increasing diameter, in each of which the steam expands and the pressure decreases, so that with a nozzle of proper shape and length the steam will leave the discharge end of the nozzle with a pressure equal to the exhaust pressure and containing the whole of the static energy of the steam in the form of kinetic energy.

The steam leaving the nozzles at exhaust pressure, there will be no leakage of the steam between the nozzles and the runner and around the runner to the exhaust and, as the runner revolves in the exhaust pressure, the friction between runner and steam will be reduced to a minimum.

The principal working part of the turbine is the runner, which is shown in Fig. 12, together with four nozzles in their proper positions. For best efficiency the angle of actual discharge from the runner-buckets, marked δ in Fig. 10, should be 90°. However, this is impracticable with the high speed of the steam obtained with a diverging nozzle, which may reach nearly 4000 ft. per second, as the rim speed would be too high and in practice the angle of actual discharge is therefore made considerably smaller, thus sacrificing a small part of the kinetic energy. The initial and terminal angles of the runner-vanes, marked β and γ in Fig. 10, are made the same and the end thrust is therefore eliminated.

The runner-disk increases in thickness from the outer rim towards the center, as by this form a much greater resistance against bursting is obtained.

The runner-shaft is very thin, and by deflecting will allow the runner to revolve about its axis of gravity. The high rotative speed of the runner, which varies between 10,000 revolutions for the larger sizes and 30,000 revolutions per minute for the smaller sizes of turbines, is reduced by means of double helical spur-gears, shown at J and K in Fig. 13, the reduction being about 10 to 1.

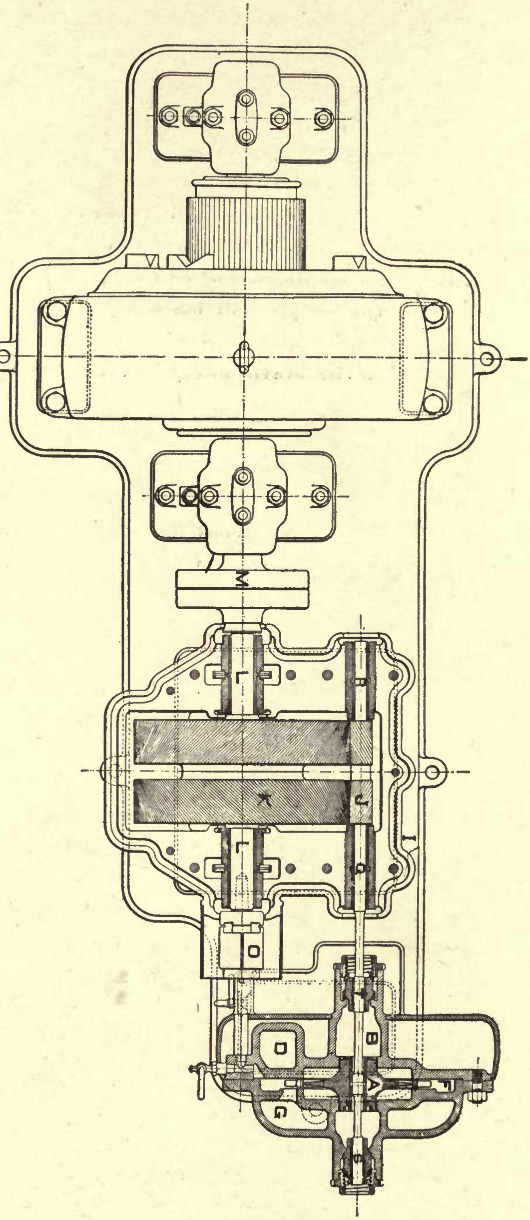

Fig. 13.—Horizontal Section of De Laval Steam turbine, with Direct-connected Dynamo. Built by the De Laval Steam Tu-bine Co., Tren ton, N. J.

The gears, or for the larger sizes the gear-rims, are of cast steel with very fine cut teeth. The helical form of the teeth not only insures smooth running, but also prevents end motion of the runner-shaft.

The governor of the De Laval turbine effects the speed regulation by the throttling of the steam, which of course somewhat reduces the efficiency at part loads. However, each nozzle is provided with a separate stop-valve operated by hand, so that when running with, say, one half of the maximum load and closing one half of the number of nozzles, the turbine will show practically the full-load efficiency.

The Rateau steam-turbine is essentially a series of action turbines of the De Laval type, all the runners being mounted upon the same shaft, like those of the Westinghouse-Parsons turbine.[1] Each runner revolves in a compartment by itself, the compartments being separated from each other by stationary disks, each compartment with its runner forming a stage in the expansion of the steam, thus reducing the discharge velocity of the steam for each individual stage.

Instead of the nozzles of the De Laval turbine, guide-buckets arranged in groups are used, which effect the passage of the steam through the stationary disks.

Mr. Rateau has also constructed a successful impulse steam-turbine with buckets of the Pelton type.

The Curtis steam-turbine is an action turbine with axial flow and partial feed, having a number of runners mounted upon the same shaft, which shaft is usually arranged vertically and carried by an oil step-bearing, that is, floating on oil under pressure.[2]

The steam is expanded in two or more stages. The guide-buckets or nozzles are diverging, as shown in Fig. 14, and for

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[1] For more detailed description see Stodola, "Steam Turbines," p. 258; also Prof. A. Rateau's paper, "Different Applications of Steam Turbines," read before the Am. Soc. M. E., June 2, 1904, and printed in abstract in Engineering News, June 9, 1904, p. 544.

[2] See Stodola, "Steam Turbines," p. 246; also Mr. W. L. R. Emmet's paper, "The Steam Turbine in Modern Engineering," read before the Am. Soc. M. E., June 2, 1904, and printed in abstract in Engineering News, June 9, 1904, p. 552.

the first stage are arranged in one to three groups, but for the following stages they may extend around the whole periphery. The velocity and kinetic energy of the steam leaving the guide-buckets is not absorbed by a single runner, but a part of it by each of the three or more runners in each stage, thus permitting

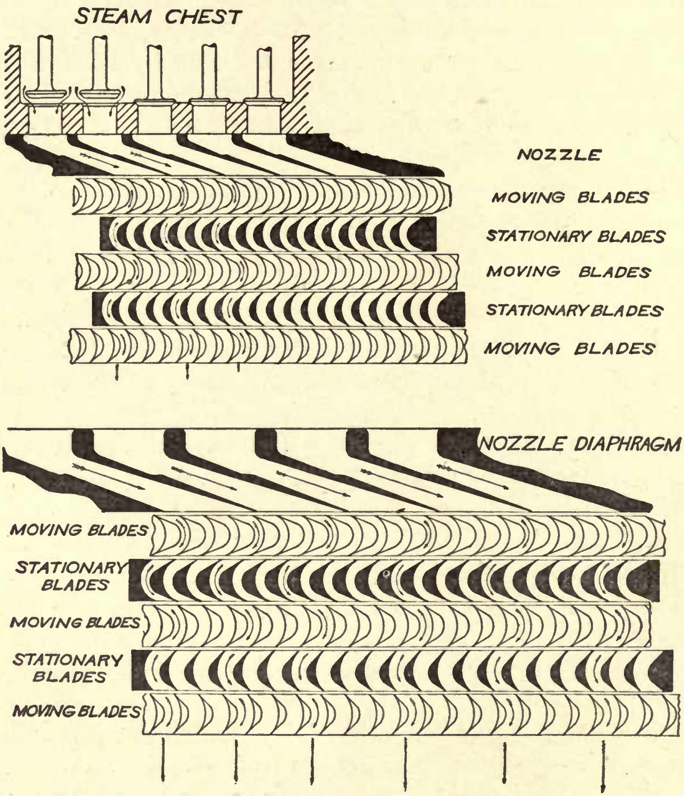

FIG. 14.—Diagram of Curtis Steam-turbine, Showing Nozzles, Runner-buckets, and Deflecting-vanes for First and Second Stage. Built by the General Electric Co., Schenectady, N. Y.

the use of a lower rim speed for the runners. Between each pair of runners deflecting-vanes are located, to redirect the steam before entering the next runner, as shown in Fig. 14. After the steam has passed all the runners of a stage and all the kinetic energy has been absorbed, the steam enters the next set of diverg-

ing guide-buckets or nozzles located in the nozzle diaphragm, which is a stationary disk forming the division wall between the successive stages.

The initial and terminal angles of the runner-vanes, marked β and γ in Fig. 10, are made the same and the end thrust is therefore eliminated.

With a two-stage turbine, built to run condensing, only the first stage is used, when the turbine is exhausting against the atmosphere.

The speed regulation of the Curtis turbine is effected by a relay governor, which operates the valves controlling the flow of steam from the admission-chamber or steam-chest to the guide-buckets of the first stage, opening or closing them one after another, as may be required. If desired, the steam admission to the following stages can also be regulated.

The working principles of the steam-turbines here described may briefly be stated as follows:

Westinghouse-Parsons series reaction turbine: The steam expands both in the guide- and runner-buckets. The velocity and kinetic energy resulting from the expansion in each individual stage are absorbed by a single runner in each stage.

De Laval single-action turbine: The steam is fully expanded in a single set of guide-buckets or nozzles and the resulting velocity and kinetic energy are absorbed by a single runner.

Rateau series action turbine: The steam expands in the guide-buckets only. The velocity and kinetic energy resulting from the expansion in each individual stage are absorbed by a single runner in each stage.

Curtis series action turbine: The steam expands in the guide-buckets only. The velocity and kinetic energy resulting from the expansion in each individual stage are absorbed by a number of runners with intervening deflecting vanes in each stage.

The following table[1] gives a comparison of the steam-turbines

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[1] From Mr. Austin R. Dodge's paper, "Advantages of Steam Turbines for Textile Mills," read before the New England Cotton Manufacturers' Assoc., 1903; also an abstract in Engineering News, Oct. 22, 1903, p. 359.

above described. The velocities given are for turbines of 300 to 600 H.P. and are approximate only.

Type.	Number of Runners.	Steam Velocity, Feet per Second.	Revolutions per Minute.	Rim, Speed.	Buckets.
Westinghouse-Parsons......	35	400	3,600	200	inserted
De Laval.................	1	4000	20,000	1200	inserted
Rateau..................	25	800	2,400	400	inserted
Curtis..................	8	2000	1,800	400	solid

CHAPTER V.

MODERN TURBINE TYPES AND THEIR CONSTRUCTION.

Turbine Construction in General.—Taking into consideration all the faults of the present turbine practice, the increased use of turbines in general, and of turbines for driving dynamo machines in particular, the necessity of radical improvements will be apparent. The writer would therefore suggest that turbines should be designed and built with the same care that is given to other high-grade machinery, and further, that three distinct types of turbines be adopted. The type to be employed in each individual case should be in accordance with the height of the head to be utilized as follows:

1. Low heads, say up to 40 ft.: American type of turbine with horizontal or vertical shaft in open flume or case, nearly always with draft-tube.

2. Medium heads, say from 40 to 300 or 400 ft.: Radial inward-flow reaction turbine with horizontal shaft and concentric or spiral cast-iron case with draft-tube.

3. High heads, say above 300 or 400 ft.: Impulse turbine or radial outward-flow full or partial action turbine, or a combination of both with horizontal shaft and cast- or wrought-iron case, often with draft-tube.

Extremes in speed or power or both will, of course, often demand the use of a turbine type for a head, outside of the range for which the type is here proposed.

Turbines with horizontal shafts should be employed in all cases, except where the use of turbines with vertical shafts is either imperative or gives a decided advantage over turbines on horizontal shafts. Horizontal turbines are not only more convenient in attendance and easier of access for adjustment or

repairs, but most of the transmission of power is done by horizontal shafts and nearly all standard patterns of direct-driven dynamos or other machinery are arranged to connect to a horizontal driving-shaft. With a very low total head or pressure-head above the turbine, it will often be necessary to use vertical turbines to be able to utilize such a head at all. In many locations horizontal turbines, on account of the great rise of the tailwater during times of flood, would have to be set at so great a height above low tailwater that the head below the turbine would be utterly beyond the practical working limit of a draft-tube during the low-water season and part of the head would thus have to be sacrificed just at the time of least water. In such a case vertical turbines are of great advantage, as they may be set at any elevation, because their being submerged during times of flood does not interfere with their operation.

The mechanical or friction losses in vertical turbines, arranged to carry the revolving parts on the water or by water thrust, are less than in horizontal turbines.

Cast-iron or steel turbine cases and draft-tubes may in most instances be dispensed with in connection with vertical turbines by making the concrete of the power-house foundations form the case and draft-tube. (See Figs. 19 and 21.)

The use of dynamos with vertical shafts direct connected to vertical turbines gives an excellent, compact, and neat arrangement, as shown by the large plants of the Niagara Falls Power Co., the German plant at Rheinfelden, the Swiss plants at Neuhausen, Beznau (Fig. 19), Chèvres (Fig. 21), the French plant at Lyons (Fig. 20), and many others.

This arrangement was also proposed by the writer for the development at Shawinigan Falls, Que., and it is to be regretted that it was not adopted, as may easily be seen by comparing the writer's proposed plan shown in Figs. 15 and 16 with the plan actually carried out, shown in Figs. 17 and 18.[1]

Dynamos with vertical shafts, having to be built specially, will cost more than standard machines, but if specially designed

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[1] The horizontal turbines installed at this plant are shown in Figs. 57 and 58.

and built dynamos are to be used in any case, the dynamos with vertical shaft will cost about the same or less than machines with horizontal shaft of the same capacity and number of revolutions.

Centrifugal pumps with vertical shafts and piston-pumps and compressors with vertical crank-shafts are also built occasionally to permit direct connection with vertical turbines where conditions make the employment of the latter imperative. (See Figs. 1 and 2.)

To transmit the power of turbines by gears should always be avoided, especially the common plan of driving a horizontal shaft from a vertical turbine by bevel-gears.

This arrangement is now much used in connection with very low heads where one or more vertical slow-speed turbines drive a horizontal shaft which runs, as a rule, at a much higher speed and is often direct connected to a dynamo or other machine. European engineers prefer direct connection of dynamos even if the dynamos have to be much larger and more expensive. Thus the power-plant at Rheinfelden, Germany, utilizing a head of from 10.5 to 14.75 ft., contains twenty direct-connected vertical units of 840 H.P. each. Nine of these units, comprising the first installation, make 55 revolutions per minute, while the remaining 11 units, using a later type of turbines, run at 68 revolutions. Another slow-speed power-plant is the one at Beznau, Switzerland (Fig. 19), utilizing a head of 12.8 to 18.7 ft. and containing 11 direct-connected vertical units of 1000 H.P. each, running at $66\frac{2}{3}$ revolutions.

The American engineer does not look favorably at slow-moving machines, especially if these machines are dynamos, on account of their greater cost, but on the other hand a slow-moving machine has a far longer life than a fast-moving one, and for direct-connected dynamos an increased speed means smaller turbine diameters and an increased number of turbines to each unit, therefore a greater cost of a turbine installation for the same capacity.

In general it may be said that turbines, both those on horizontal and on vertical shafts, should be self-contained; that is, all stationary parts should be securely connected to one rigid main frame or base, or to such main parts of the turbine as are

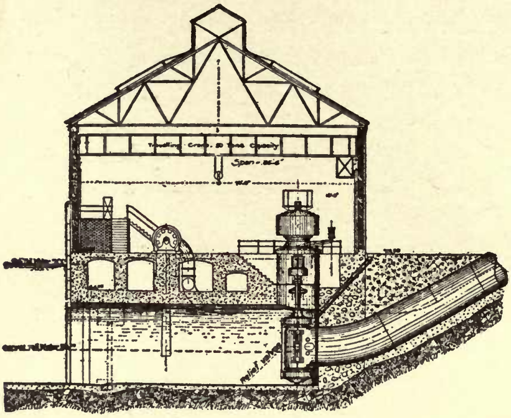


FIG. 15.

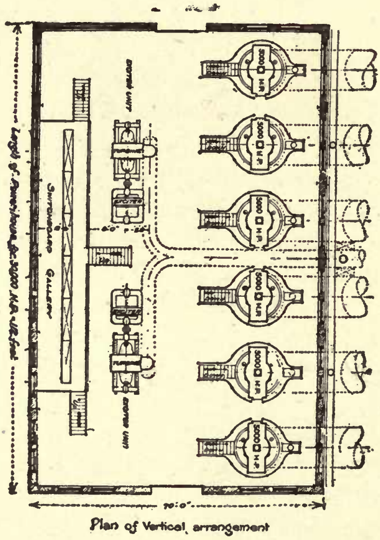


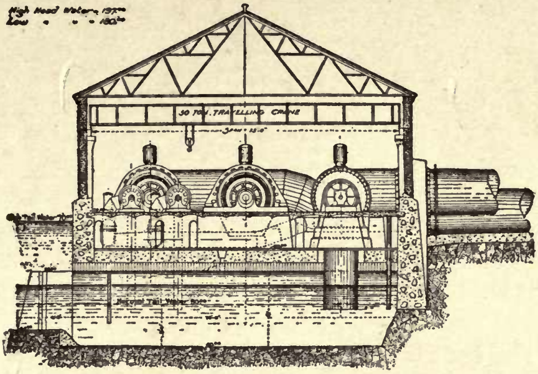
FIG. 16.

FIGS. 15 and 16.—Proposed Vertical Arrangement of Turbines for the Power-house of the Shawinigan Water and Power Co., Shawinigan Falls, Que.

Area occupied by power-house: Horizontal, 18,400 sq. ft.; vertical, 7840 sq. ft.

Increased cost of horizontal over vertical arrangement:

On excavation, at $1 per cubic yard.	$9,740
" concrete and floors, at $8 per cubic yard.	40,000
" penstocks, at $55 per running foot.	10,000
" building, brick, roof, windows, and heating-plant. .	5,420
	$65,160



Cross Section
Horizontal arrangement

Fig. 17.

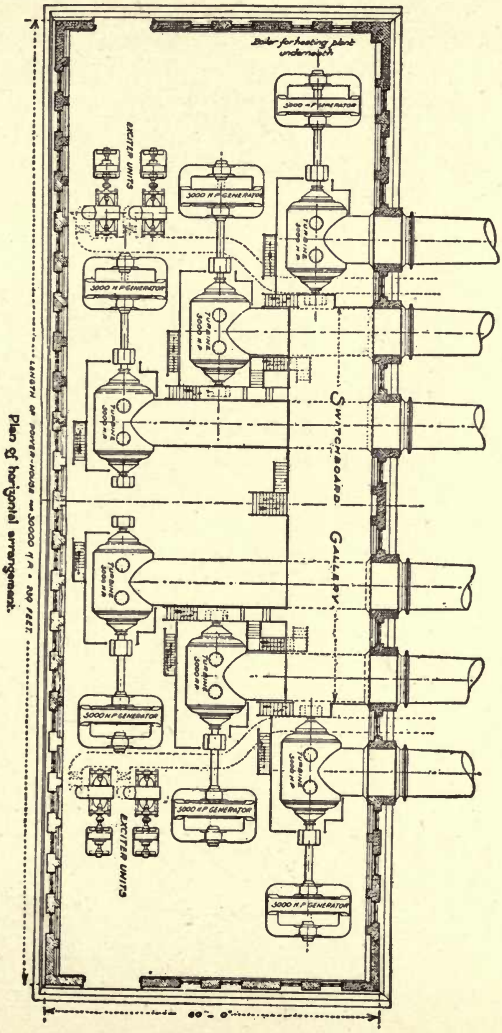


Fig. 18.

Figs. 17 and 18.—Horizontal Arrangement of Turbines for the Power-house of the Shawinigan Water Power Co., Shawinigan Falls, Que., as Actually Installed. Designed by W. C. Johnson, C.E., Montreal, Que.

arranged to serve as a frame or base, but a turbine case of iron or steel plate should never be employed as a frame.

Main frames or bases should be used in all cases, even where the turbine rests on solid masonry or concrete work, and they should be constructed of cast iron in preference to wrought iron or wrought steel, as cast iron is more rigid than the wrought metal. The shape of the frame will vary according to the general arrangement of the turbines, but a form of box girder with deep webs, well-stiffened sideways, and with wide flanges will answer in most cases. The joints between the frame parts themselves and between the frame and the turbines should be planed.

Turbine frames should never be made to serve as beams, and where turbine-pits or tailraces are directly below the turbines, masonry arches or concrete and steel contruction should be employed to bridge over these openings and to support the turbine-frames.

Where turbine-shafts are direct connected to the shaft of dynamos or other machinery, it is advisable to have the main frames of the turbines extended or separate connecting pieces provided, so that the frames of the turbines and those of the dynamos or other machinery may be bolted together rigidly, thus making the alinement and proper working of the connected machines independent of careless and faulty erection and unequal settlement.

Turbine-cases should be of such shape that all changes in speed and direction of the water are made gradually. Water, on account of its inertia, cannot change its speed instantly, and therefore any sudden change in the clear area of a water conductor will form a pocket in which part of the main flow of the water is detained and whirled around.

All turbine-cases should be provided with proper drain-valves to empty them of water or to flush out stones, sand, etc.

To admit the water to the turbine-case at the end or in an axial direction should be avoided, as this arrangement implies that one of the main shaft-bearings is to be located in the water. With turbines set in an open chamber, it can rarely be avoided to have one of the main bearings under water.

Where circumstances will permit, a case with end inlet similar to the type shown in Fig. 35 may be used, which could easily

be modified so as to have both main bearings outside of the case.

Except where required to carry the weight of the revolving parts, it is to be avoided to have the full head or water pressure acting against the runner-disk of reaction turbines. The pressure against the disk, due to the leakage from the clearance, cannot be avoided, but may generally be relieved and the end thrust thus reduced by having openings in the runner-disk connecting the space between the disk and the head of the case or the gate dome with the discharge side of the turbine. (See Figs. 25, 26, 29, 30, 34, and 35.)

The American type of runner, being a very difficult piece to mold, is now often built up from separate pieces, each piece forming one or more buckets, but such built-up runners are only permissible if they are very carefully made and strongly banded.

Manufacturers making the runners of the American type in one casting frequently use a very soft cast iron, as such iron when molten is more fluid and better fills the molds than the harder metal. This means a rapid wear of the buckets, due to the erosion caused by the sand and gritty matter carried by the water. The writer has in mind an instance where 48-in. turbines, working under a head of only 28 ft., wore so rapidly that after three years of operation nothing remained of some of the runners but the hubs and small fragments of the buckets.

As the erosion may be taken to increase directly with the head or with the square of the velocity of the water, it will be evident that for high heads, especially as mostly action turbines are used in connection with them, the buckets should be of a hard cast-iron, steel casting, manganese, or phosphor-bronze to reduce the wear to a minimum. Of course, with high heads and small volumes of water, that is with small rates of flow, much better precautions can be and are usually taken to free the water from sand and gritty matter than is the case with low heads and large volumes of water.

To free large volumes of water from sand and gritty matter is often very difficult and expensive, and in such instances it will be found cheaper, as a rule, not to attempt it, but to renew guide- and runner-buckets when worn out. Except for turbines of very

small diameter, the guide- and runner-vanes with their crowns should therefore be cast as separate rings, to be bolted to the turbine-case and the runner-disk respectively, as this will greatly lessen the cost of renewal, besides making these parts somewhat easier to mold. The threads of the bolts used for this purpose should be coated with red lead or graphite to prevent the nuts from rusting fast.

Corrosion of iron is caused by the combined action of moisture and the carbonic acid in the atmosphere. There is little trouble with corrosion of the buckets of reaction turbines which work continuously or are always submerged, but the backs of the runner-vanes of action turbines with free deviation and of the buckets of impulse turbines will often corrode and pit very quickly, as they are always in contact with the air and continually wetted by splashing water, thus presenting the most favorable conditions for rapid corrosion.

European builders therefore now frequently make the runner-buckets of action and impulse turbines working under high heads of manganese bronze, as this material not only offers great resistance to corrosion and erosion, but also has a great strength, its ultimate tensile strength varying between 45,000 to 60,000 lbs. per sq. in. for the cast metal, with an elongation of 8 to 12%. Phosphor-bronze having an ultimate tensile strength of from 44,000 to 52,000 lbs. per sq. in. and an elongation of 8 to 33%, is also often used for this purpose.

Should a governor regulating the speed of a turbine fail to act while the turbine is running without a load, the speed of the turbine may increase to almost twice the normal speed, and the runner should for this reason be capable of withstanding twice the normal speed without being in immediate danger of bursting.

The end thrust in a plain inflow reaction turbine is principally due to the water-pressure between the head of the case or the dome and the runner-disk. To this has to be added for turbines in which the direction of the water is changed from the radial inward to the axial direction, while flowing through the runner-buckets, as in the vortex turbine, the axial water-pressure against the discharge end of the runner-vanes, for it is this part of the vanes which deflects the water from a more or less axial

direction to the realtive direction of discharge denoted by c_t in Fig. 10. Opposed to the end thrust is the pressure, due to the deflection of the water, from the inward to the axial direction while flowing through the runner-buckets.

End-thrust bearings with wooden steps should be abandoned and metal steps, or, better still, metal collar thrust-bearings, used instead. Such metal thrust-bearings should, of course, never be located in the water, but are usually placed on the end of the shaft, opposite to the end from which the power of the turbine is taken off. Both the straight and the collar bearings of the main turbine-shaft should be adjustable, and should be lined with bronze as a base, and the bronze in turn lined with an anti-friction metal or babbitt well-hammered and bored.

For turbines having a great end thrust, such as single turbines working under a head of several hundred feet, the thrust-chamber should be employed. Turbines having runners of such size or shape as to prevent the use of a thrust-chamber, as the American or vortex turbine for example, should employ the thrust-piston instead, placing the runner at one end and the piston at the other end of the case.

No packing is used for a thrust-piston, but the piston closely fits the cylinder, and grooves are turned in the piston, as shown in Fig. 5, to reduce the leakage of water.

If the water used for a turbine contains much sand or gritty matter, a separate water-supply should be provided for the thrust-piston, and this water should be run through a sand-settler or a filter or screen, to free it from the sand or gritty matter, as otherwise the thrust-piston will wear rapidly.

Turbines having a thrust chamber or piston or double turbines in which the two end thrusts balance each other, always require in addition a small collar-bearing, to take care of unavoidable variations in the end thrust.

Cylinder gates are the simplest form of gate arrangement, but their use considerably increases the axial dimension of the whole turbine, and the distance between main bearings has to be much greater than is required with register or wicket gates.

To have bolts, nuts, lugs, or other projections in the water-passages of the guide-buckets, as now frequently found in con-

nection with wicket gates, is strongly to be condemned, as such bolts, nuts, or lugs prevent the smooth flow of the water and produce friction and eddies, thus greatly reducing the efficiency.

By using register or wicket gates for double turbines the center bearing may be dispensed with.

All outside journal-bearings should be self-oiling, such as ring or chain oiling bearings.

For all journals located under water, for which wooden bearings are now employed, metallic bearings should be used provided with forced oil circulation, as shown in Fig. 25.

Such oil circulation has also the advantage that the oil returning from the bearings will show whether any cutting takes place in the bearings.

The oil force-pumps should always be installed in duplicate, so that the breaking down of an oil-pump will not necessitate the stopping of the whole plant.

The body of the stuffing-boxes for the main shaft should be separate from the heads of the turbine-case and bolted to the heads, so that by loosening these bolts their centers may be slightly shifted. This will permit the runner to be accurately centered with the guide-ring and the main bearings to be properly adjusted, moving the stuffing-boxes with the shaft as required. Such a shifting arrangement for the stuffing-boxes is of great importance in connection with turbine-cases made of steel plate, as these cases are likely to spring or warp considerably.

The mechanism for actuating the gates should always be outside the turbine case or flume and connected to the gates themselves by rods, entering the case or flume through stuffing-boxes. Gears for moving the gates should be avoided, and levers, links, or hydraulic pistons, or a combination of these, used instead.

To have everything, even the gate domes and draft-tube tees, inside a large and all-surrounding case, as is now the general practice for the sake of cheapness, is not only unnecessary but must be regarded as an unmitigated nuisance.

Most gates have a strong tendency either to close or to open, according to their arrangement, and this force always tending to move the gates in one direction should be counterbalanced, especially when a governor is employed for regulating the gate

movement. However, this balancing should not be done by counterweights and chains, but by the pressure of the power-water, applied to parts of the gates provided for that purpose or to balancing-pistons. Besides doing away with weight, chains, and their rigging, such an arrangement has the advantage that any variation in head which increases or decreases the tendency of the gate to move also changes the balancing pressure in the same proportion. Where a long penstock is used to supply turbines having gates with a tendency to close, as is the case with nearly all gates, it is advisable to have these gates provided with dashpots to prevent them from moving too rapidly, as otherwise the penstock may be wrecked should any part of the gate rigging give way and the gates thus be permitted to close suddenly.

The turbine-governor should be regarded as an essential part of the turbine, and for horizontal turbines at least the governor should be mounted either on the turbine frame or on the case. The governor-drive should be positive, that is by shafts and gears in preference to a belt. The gears used for the governor-drive and such gears as may be found in the governor itself should be the only gears employed about a turbine unit.

The careful design of turbines, mentioned above as necessary, includes, of course, easy access to every part and at all stages of the water.

Proper manholes should therefore be provided and shafts, runners, and guide-rings arranged, so that they may be easily removed without dismantling the whole turbine.

All anchor-bolts and similar parts should be arranged to permit of easy renewal. Bolt-threads should not be located in damp or wet places, but where this cannot be avoided, the threads should be coated with red lead or graphite to prevent the nuts from rusting fast, and after the nuts have been tightened, the bolt ends, nuts, and washers should receive a thick coat of asphaltum, to exclude the moisture.

Turbines for Low Heads.—For low heads the writer considers the American turbine as the best type, and would recommend that the turbines be set above tailwater and supplied with draft-tubes in all cases where the head is sufficient to give enough depth of water above the turbine, so that the buckets will always be

properly filled and the air prevented from being sucked through.

Where a sufficient depth of water above the guide-buckets cannot be secured or other reasons make it advisable to install vertical turbines, it has been the univerasl practice in America to use single vertical turbines, and as such turbines under the low heads for which they are employed give, as a rule, too low a speed if of the proper horse-power, or too low a horse-power if running at the proper speed, the gearing of one or more vertical turbines to a horizontal shaft becomes necessary.

As far as the writer is aware, multiple turbines on vertical shafts and working under low heads have never been used in American plants of any importance, while they are very common in European practice. The turbines in the power-house No. 1 of the Niagara Falls Power Co. are double turbines on vertical shafts, but they are working under a mean head of 138 ft.

However, the vertical multiple turbine is deserving of a more general application for low heads, and as their arrangement is practically unknown to most American engineers, a number of typical plants will here be illustrated, but it may be stated first, as pertaining to all such plants, that vertical turbine units which are continually or at times of flood-water partly or wholly submerged must each be placed in an independent tailrace. Each tailrace has to be provided with slides for stop-logs or a gate and connected by piping with a centrifugal pump, so that by putting in the stop-logs or closing the gate, opening the proper pipe connections and starting the pump, any tailrace may be emptied and the turbine unit made accessible for inspection or repairs. (See Figs. 15, 19, and 21.) One gate will serve for all tailraces. The slides are usually located at the mouth or outlet of the tailrace, but it is better, especially in cold climates, where the formation of ice in the slides would make it difficult to insert the stop-logs or gate, to have the slides inside of the power-house foundation and to let the stop-logs or gate down into the tailrace through slots in the power-house floor, which slots are ordinarily closed by cast-iron plates set into the floor and only opened when required to admit the stop-logs or gate. With this arrangement no extra hoisting apparatus is necessary, as the traveling-crane can handle stop-logs or gate.

Another rule to be observed in connection with vertical turbine units is that the weight of the revolving parts, or at least a portion of that weight, should always be carried by the water or by water thrust to relieve the collar-bearing. Where it is desired to have the same portion of the weight carried by the water at all stages of the headwater, the thrust-chamber or pis-

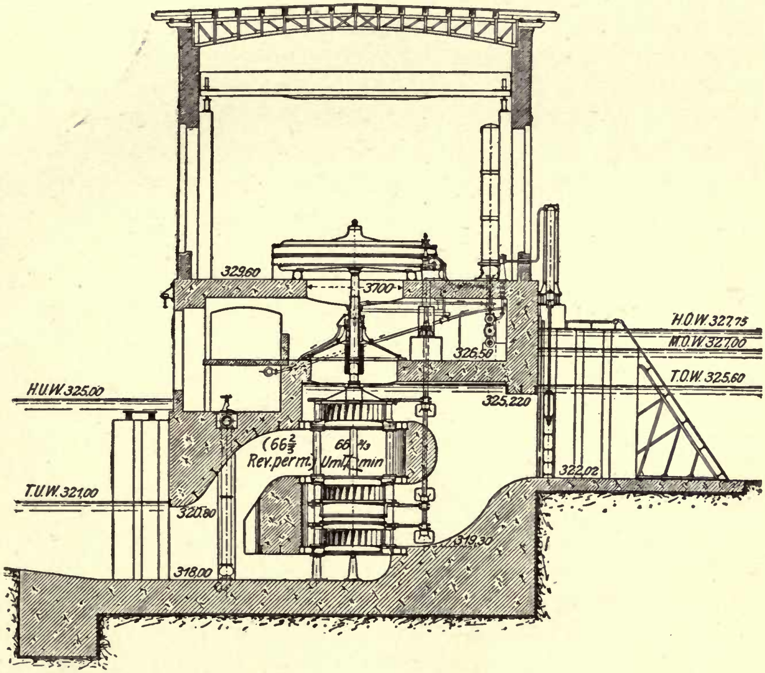


FIG. 19.—Cross-section of Power-house of the Elektrizitaetswerk Beznau, Beznau, Switzerland. Turbines built by Th. Bell & Co., Kriens, Switzerland.

ton should be of such size as to carry the desired weight at the lowest stage of the headwater and the water should be supplied to the thrust chamber or piston by a separate pipe, the water in this pipe being always kept at a pressure corresponding to the lowest stage of the headwater by means of a reducing-valve.

In Fig. 19 is shown a cross-section of the power-house of the Elektrizitaetswerk Beznau, at Beznau, Switzerland, built in 1900

and 1901. The plant contains eleven main turbine units, each unit being a vertical triple turbine developing 1000 H.P. under a head of from 12.8 to 18.7 ft. and running at 66⅔ revolutions. The turbines are of the inflow reaction type, and to attain even the low speed at which they are running a very high reaction had to be used, giving a speed factor of 0.925. The speed regulation is effected by gates similar to those shown in Figs. 40 and 41, moving simultaneously in the three turbines of each unit. The end thrust of the top and bottom turbines is partly relieved by openings in the runner-disks, while the water-pressure against the under side of the solid disk of the middle turbine carries a portion of the weight of the revolving parts. As shown in the illustration, the turbine-cases and draft-tubes are formed by the concrete foundations of the power-house.

In Fig. 20 is shown a cross-section of the power-house of the city of Lyons, near Lyons, France. The plant contains eight main turbine units of 1250 H.P. each, comprising the first installation, and eight main units of a later design of 1500 H.P. each, all working under a head of from 26.3 to 37.6 ft. and running at 120 revolutions. The turbines seen in the illustration is a 1250-H.P. unit and is a single three-story inflow reaction turbine having a runner of conical shape, with the smaller diameter at the top. The bottom and middle story of the runner are used at all stages of the water, while the top one is used in addition during flood-water, when the head is the lowest. The speed regulation is effected by a separate cylinder gate for each story, located at the entrance or outside of the guide-bucket rings and moving simultaneously, but the gate for the top story is disconnected from the governor when that story is not in use. The unit is inclosed in a cast-iron case in the upper head of which a thrust-piston is located, carrying 22 tons of the weight of the revolving parts.

In Fig. 21 is shown a cross-section of the power-house of the city of Geneva, at Chèvres, Switzerland. This plant contains five main turbine units of 1000 H.P. each and running at 80 revolutions, comprising the first installation, and ten main units of a later design, each of 1200 H.P. and running at 120 revolutions, all turbines working under a head of from 14.1 to 26.2 ft. The

turbine seen in the illustration is a 1200-H.P. unit and is a vertical quadruple turbine of the outflow reaction type. The runners are arranged in pairs, the lower pair alone being used when the

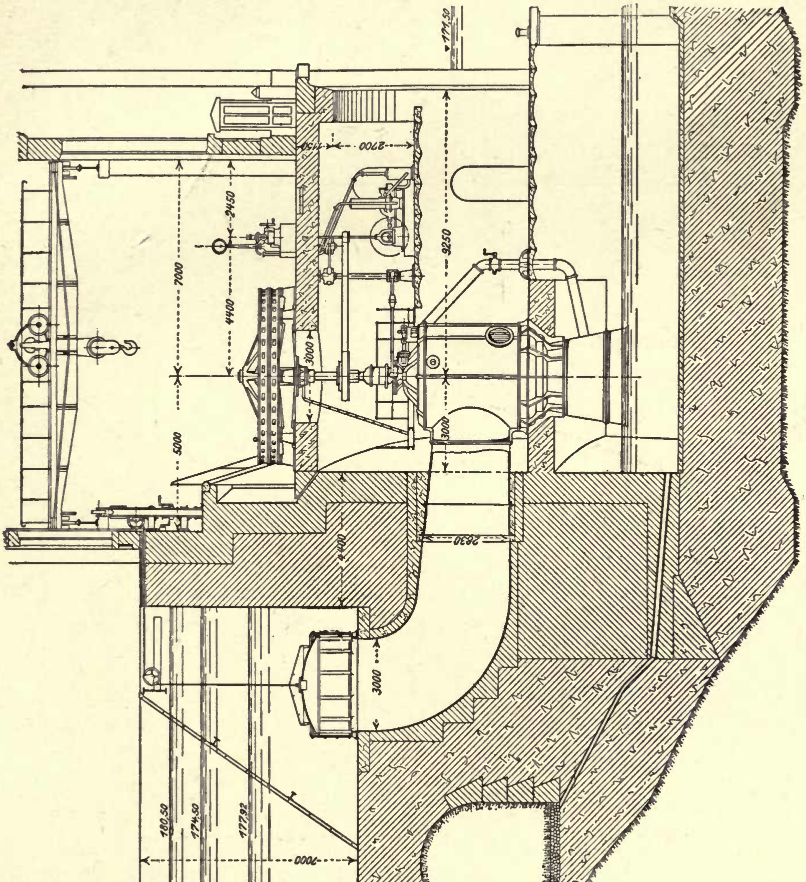


FIG. 20.—Cross-section of Power-house of the City of Lyons, near Lyons, France. Turbines built by Escher, Wyss & Co., Zurich, Switzerland.

head is at or near its maximum. Each runner has a separate cylinder gate, located at the discharge or outside of the runners and moving simultaneously, but the gates for the upper pair are disconnected from the governor when that pair is not in use. Each pair has only one disk in common for the two runners, and

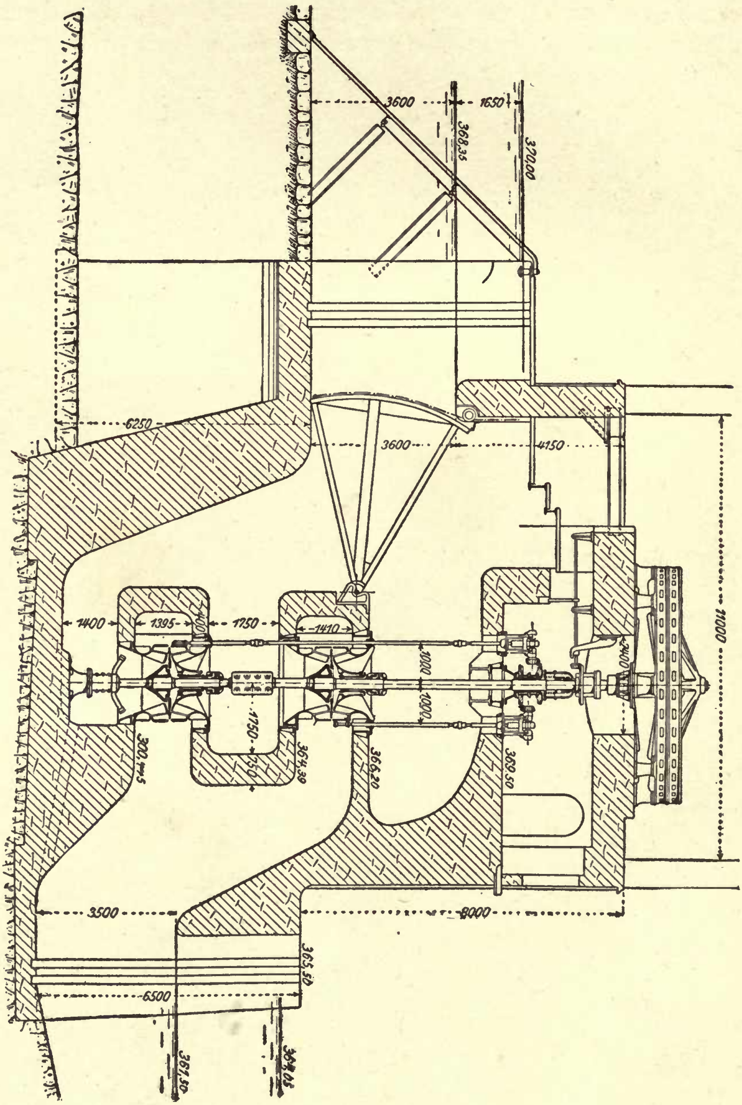


Fig. 21.—Cross-section of Power-house of the City of Geneva, at Chèvres, Switzerland. Turbines built by Escher, Wyss & Co., Zurich, Switzerland.

the disk being solid, the water-pressure against its under side carries nearly the whole weight of the revolving parts. The water-pressure is prevented from acting against the upper side of the runner-disks by making the disk of the upper guide-bucket ring of each pair of turbines solid and carrying the shaft through by means of a stuffing-box.

As shown in the illustration, the turbine-cases and draft-tubes are formed by the concrete foundations of the power-house.

The unusual head-gates seen in Figs. 20 and 21 will be considered under "Head-gates."

The turbines recently installed in the electric works on the river Glommen in Norway and here shown in Figs. 22 and 23 are interesting examples of single inflow reaction turbines with vertical shafts.[1] Vertical turbines were adopted for this plant which will have an ultimate capacity of 50,000 H.P., on account of the great variation in the tailwater, the floodwater rising to 40.5 ft. above the minimum tailwater level. To avoid the use of multiple turbines to obtain the required horse-power under the available head and at the desired number of revolutions, turbines of a modified American or vortex type were selected. The turbines work under a head of from 39.6 to 63.1 ft., develop 3000 H.P. each with a head of 52.5 ft. and run at 150 revolutions. The speed regulation is effected by wicket gates. The weight of the revolving parts is 32 tons and is carried by a collar-bearing, which is relieved by oil under pressure; the pump furnishing this oil also operates the relays of the speed-governors. The turbine-case is spiral in form, made of steel plate, and surrounds the guide-ring only. The draft-tube is formed in the concrete of the power-house foundation, and an egg-shaped sewer is located above the draft-tube outlets, to collect any water which may leak into the basement or turbine story during high water, and a centrifugal pump is provided to empty the sewer when required.

The best arrangement for a single or multiple horizontal turbine with draft-tube or tubes is an open turbine-chamber, built of masonry or concrete or concrete and steel and forming a direct

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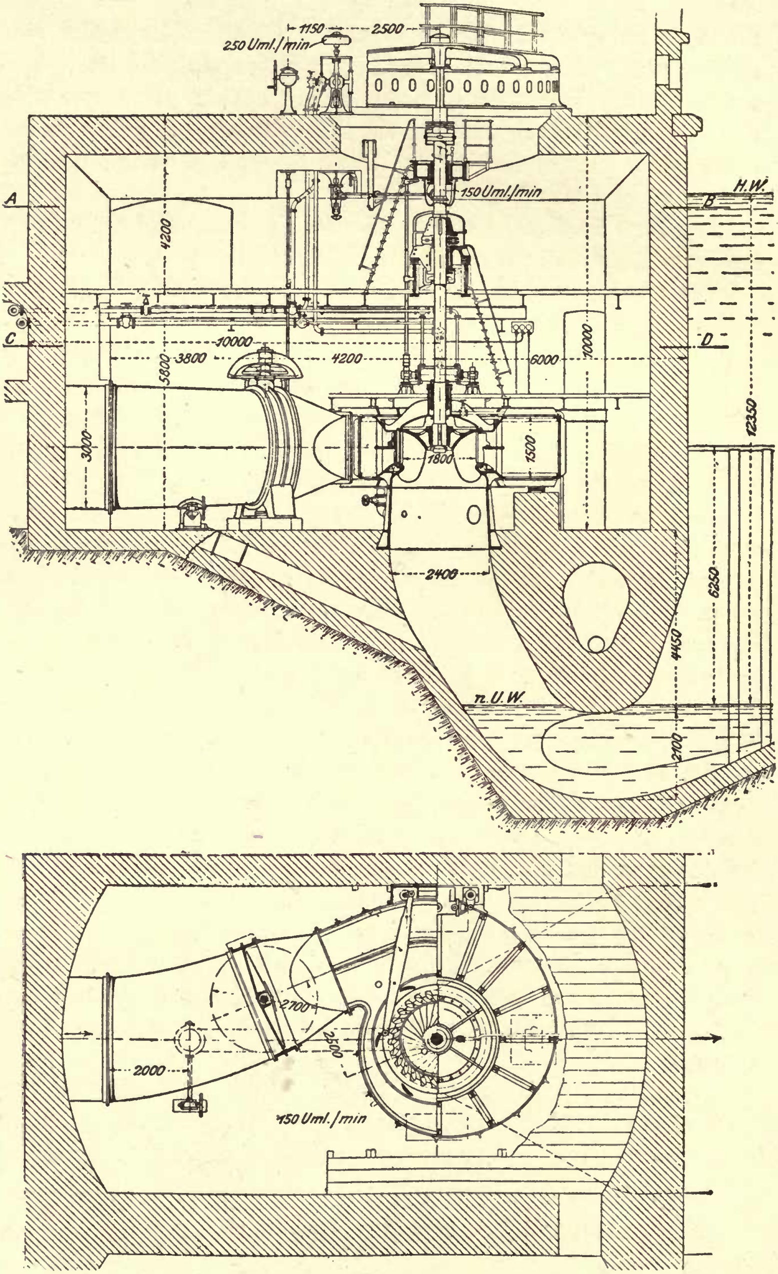
[1] Zeitsch. d. V. deutsch. Ing., June 20, 1903, p. 891.



FIGS. 22 and 23.—3000-H.P. Turbine for the Electric Works on the River Glommen, Norway. Built by J. M. Voith, Heidenheim, Germany.

continuation or branch of the headrace or forebay. This plan has been adopted in recent years for many important power-plants, of which may be mentioned here the plants of the Hudson River Power Transmission Co., Mechanicsville, N. Y.; the St. Lawrence Power Co., Massena, N. Y.; the Michigan Lake Superior Power Co., Sault Ste. Marie, Mich.; the St. Anthony Falls Water Power Co., Minneapolis, Minn.; the Montreal Light, Heat, and Power Co., Montreal, Que., etc.

A cross-section of the power-house of the last-named company, located at Chambly, Que., is shown in Fig. 24. As planned this plant was to contain eight main turbine units of 2648 H.P. each, running at 153 revolutions per minute and working under a head of 28 ft., obtained by damming the Richelieu River.

Each turbine unit was direct connected to an alternating dynamo and consisted of four 48-in. turbines.

Open turbine-chambers have three advantages over turbine-cases, viz., the friction of the water flowing to the turbine is reduced to a minimum, the turbines are very convenient of access, and the chambers present the best possible conditions for the speed regulation of the turbines, a very important consideration where the water-power is used to drive dynamos.

The least permissible depth of the water above the highest point of the entrance rim of the guide-bucket ring, according to European practice, is 1 meter (3.28 ft.), but even then funnels may form and air be sucked through the turbine, which not only causes a loss in power and considerable loss in efficiency, but also produces irregularities in the speed of the turbine.[1] For general practice a minimum depth of water of 3.5 to 4.5 ft. should be allowed above the highest point of the entrance rim of the guide-ring, or 3.5 ft. for turbine-chambers in which the water flows towards the guide-rings at a high speed, say 5 to 6 ft., increasing to 4.5 ft. for chambers in which the velocity of approach is low, say 1 to 2 ft., as may often be the case, when the turbines are running with part gate.

The minimum depth of water above guide-buckets as here given applies only to turbines with a low draft-head. The influ-

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[1] Mueller. Francis-Turbinen, p. 161.

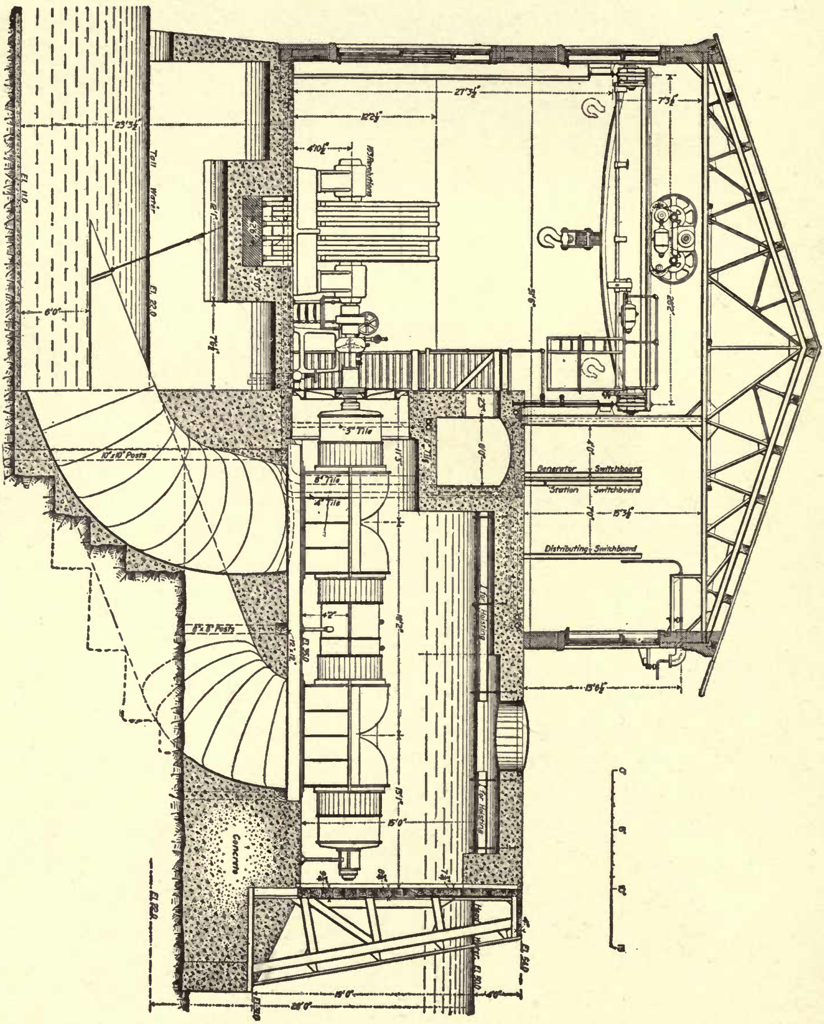

FIG. 24.—Cross-section of Power-house of the Montreal Light, Heat, and Power Co., at Chambly, Que.

ence of the draft-head on the water above the turbine will be considered under "The Draft Tube."

Where this minimum depth cannot be obtained at all stages of the water, the formation of air-funnels can be prevented by mechanically agitating the water above the entrance to the guide-buckets. Such agitators must be under water to avoid the formation of air-bubbles. As a better plan the writer would suggest to fasten, above the entrance to the guide-buckets, a metallic plate having a length and width somewhat larger than the outside diameter of the guide-ring. This plate should be horizontal, with its edge slightly turned up on the side from which the water approaches, and its location should be centrally above the guide-ring and at such a height that the lower surface is still in the water when the latter is at its lowest stage. Such a plate would act the same as the shell of a turbine-case, that is, it would seal the water against the air. A board of the same dimensions, floating on the water and properly guided, would serve the same purpose, but suitable stops would have to be provided to prevent the board from going below the low-water level, as occasionally the downward suction might prove greater than the buoyancy of the board.

Open timber flumes or timber supports for turbine settings should never be used except for temporary installations.

In Fig. 25 is shown a longitudinal section of a horizontal double turbine, built for the Isarwerk, near Munich, Germany. These units are arranged to be set in open turbine-chambers and under a head of 38 ft. are developing 2500 H.P. each and running at 150 revolutions. The turbines are of the inflow reaction type, and to obtain the required speed a high reaction had to be used, giving a speed factor of 0.82. The speed regulation is effected by register gates of the design shown in Figs. 3 and 4. The center bearing of the main shaft is dispensed with, the end bearing, located inside the chamber and therefore under water, is inclosed and provided with forced oil circulation, and the draft-tee and draft-tube are of steel plate.

Where such open flumes cannot be used, the dome or casing inclosing the gate arrangement should be strengthened to stand the water-pressure due to the head, and a concentric, or, better, a

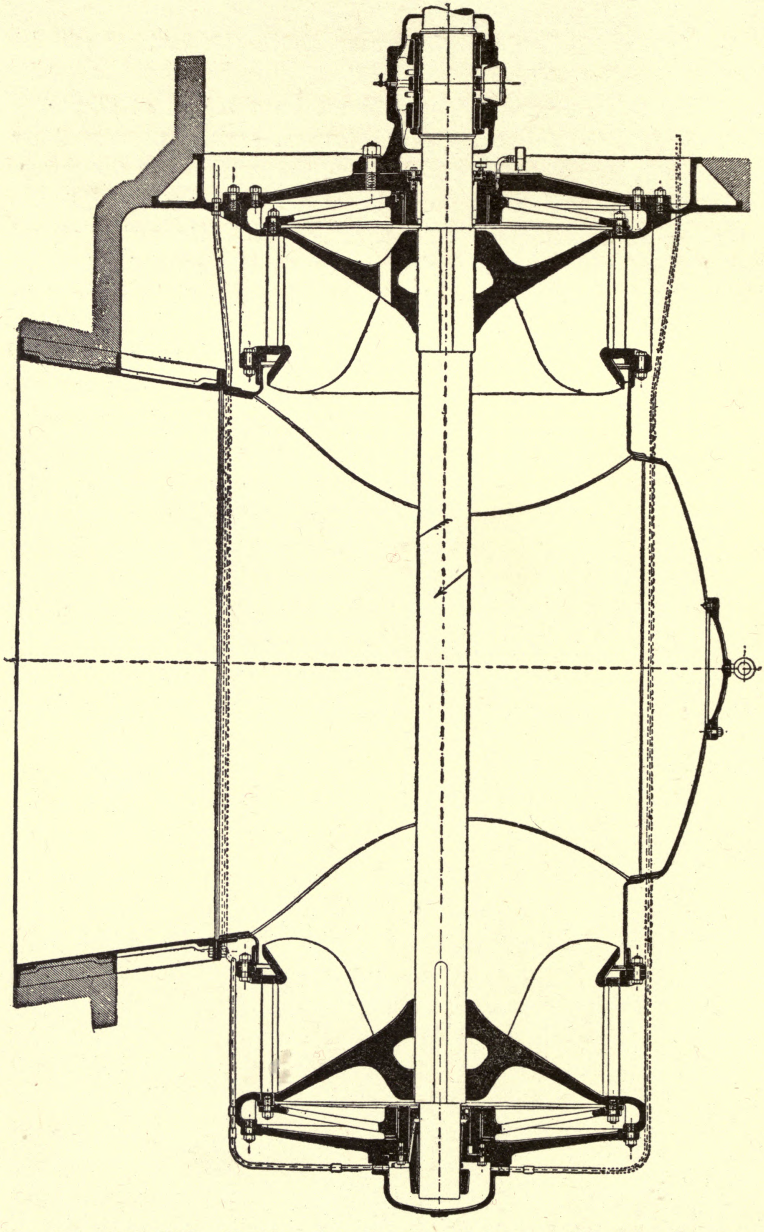

FIG 25.—2500-H.P. Turbine for the Isarwerk, near Munich, Germany. Built by Escher, Wyss & Co., Zurich, Switzerland.

spiral cast-iron case added to the turbine surrounding the guide-ring and the guide-ring only, in the same manner as the case of a centrifugal pump surrounds the fan-wheel. The Figs. 26 to 34, illustrating turbines for medium heads, show various forms of such cases. The entrance to the turbine-case should have an area not greatly in excess of the total entrance area of all the guide-buckets, which area is equal to the axial dimension of the guide-bucket entrance multiplied by the outer circumference of the guide-ring. The water while entering and inside of the case would thus have a speed nearly as high as when entering the guide-buckets. The increased friction loss, due to the higher speed of the water in the turbine-case, will be found to be less than the losses caused by the abrupt changes in the speed and the direction of the water in the ordinary steel-plate cases.

The penstock, or the penstock nozzle connected to the case, should be gradually reduced to the diameter of the entrance to the case. (See Figs. 38 and 39.)

For very low heads and large powers where the case would be of very large dimensions, or where required for other reasons, the case might be made of steel plate, shaped as nearly like a cast-iron case as can be produced conveniently. and having a somewhat larger cross-sectional area than necessary for cast iron. (See Figs. 38 and 39.)

Double turbines may be arranged to discharge towards each other, and then should have a case common to both, of a form as shown in Figs. 36 and 37, or, better, a separate case for each turbine, of a form as shown in Figs. 38 and 39. Double turbines discharging in opposite directions would best be arranged with the runner-disks set against each other and may then have not only the case but also the guide-ring in common to both turbines, as shown in Figs. 40 and 41, thus requiring a minimum of floor space.

The speed-regulating gates employed may be cylinder, register, or wicket gates, but register and wicket gates are preferable Care should be taken to have the gates well balanced.

Turbines for Medium Heads.—For medium heads the writer considers the radial inward-flow turbine in a cast-iron case as the best type. These turbines should be either reaction or limit

turbines, according to the head, and should always be provided with draft-tubes and set above the tailwater.

With medium heads the use of vertical turbines will only be required under exceptional conditions. Such conditions were met with at Niagara Falls, where the 5500-H.P. units in power-house No. 1 of the Niagara Falls Power Co. are double outflow reaction turbines on vertical shafts, working under a mean head of 138 ft. and running at 250 revolutions. These turbines have no draft-tubes and no turbine-cases.[1] The turbines in power-house No. 2 of that company, already referred to in connection with the thrust-piston and shown in Fig. 5, are single vertical inflow reaction turbines of 5500 H.P. each, working under a head of 156 ft. and running at 250 revolutions.[2] Each turbine is inclosed in an almost spherical cast-iron case and has a draft-tube, which is divided into two branches, straddling the tailrace, as a single center draft-tube would obstruct the tailrace. The speed regulation of the turbines is effected by a single-cylinder gate and the runner has no additional crowns—that is to say, is only one story high—but as there will be eleven such turbines in the power-house, a number of these can always be run at full gate, while the remaining units are started or stopped, as the demand for power may require. The weight of the revolving parts, as already stated, is carried by the thrust-piston at the lower end of the shaft.

The 10,500 H.P. units of the Canadian Niagara Power Co. are double vertical inflow reaction turbines, working under a head of 136 ft. and running at 250 revolutions. Runners and guides are practically the same as shown in Fig. 5, and the draft-tubes are similar in arrangement. The runners discharge towards each other. The cast-iron case is, of course, different from that shown in Fig. 5, and the thrust-piston is located above the upper turbine.[3]

The turbine units intended for the Shawinigan Water and

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[1] Wood. Turbines, p. 130.

[2] Zeitsch. d. V. deutsch. Ing., Aug. 31, 1901, p. 1239; also an abstract in Engineering Record, Nov. 23, 1901, p. 500.

[3] For an illustrated description of these turbines see Engineering Record, Dec. 19, 1903, p. 765.

Power Co., Shawinigan, Que., seen in the cross-section of the power-house, Fig. 15, were double vertical outward-flow reaction turbines developing 5500 H.P. each under a head of 125 ft. and running at 225 revolutions. These turbines were to have no draft-tubes or cases, as they would always have been wholly or partly submerged. Each runner was to be divided into three stories and the speed regulation effected by cylinder gates. The disk of the upper runner or a thrust-piston, the latter supplied with pressure-water by a separate pipe, were proposed to carry the weight of the revolving parts.[1]

For horizontal turbines the cases should resemble in outward appearance those of centrifugal pumps, and may be either concentric, like a Harmon pump-case (see Figs. 30 and 31), or, better, spiral, like the case of an ordinary centrifugal pump (see Figs. 26 to 29 and 32 to 34). The concentric case has the advantage that it can always be used either right or left hand, while the spiral case is smaller in size, permits of a higher speed of the water flowing through it, and gives a slightly better efficiency to the turbine.

The regulating-gates used should be either register gates (see Figs. 3, 4, 25, and 28 to 30) or wicket gates (see Figs. 26, 27, 32, 34, 40, and 41); the latter have been found to give a somewhat better efficiency for part gate-opening, but are more complicated than the register gates.

These turbines are mostly built as single turbines, and for small sizes the end thrust is taken care of by a step- or collar-bearing, while for large sizes the thrust-chamber or piston is employed.

It is often asserted that the close spacing of the buckets of European turbines causes trouble by choking of the buckets with bark, frazil, anchor ice, etc., but experience in Europe has shown that this is not the case, provided the racks are kept in proper repair, and turbines having only 1 in. clear space between the vanes forming the guide- and runner-buckets are now running in Norway without the least trouble from bark or ice.

In Figs. 26 and 27 is shown a small single turbine in spiral cast-iron case and intended to develop 104 H.P. under a head

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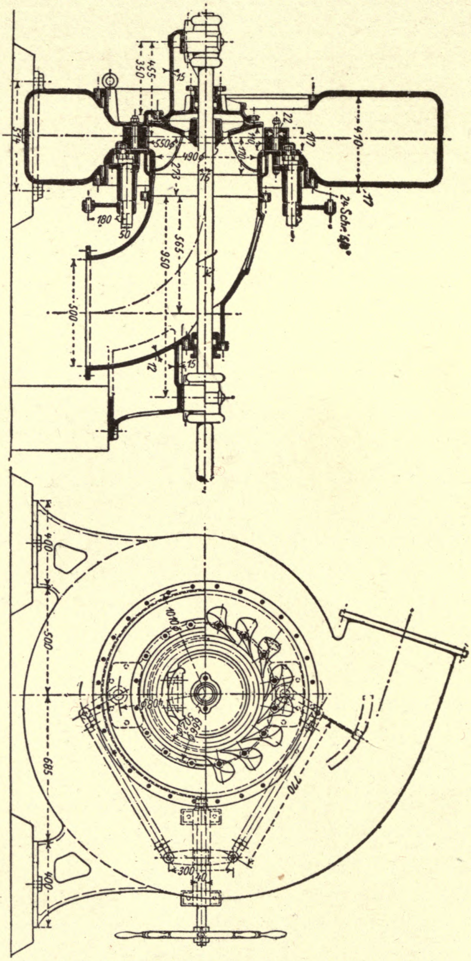
[1] See Canadian Engineer, May 1901.

of 49 ft. and to run at 374 revolutions. The wicket gates are a modification of Fink's design.

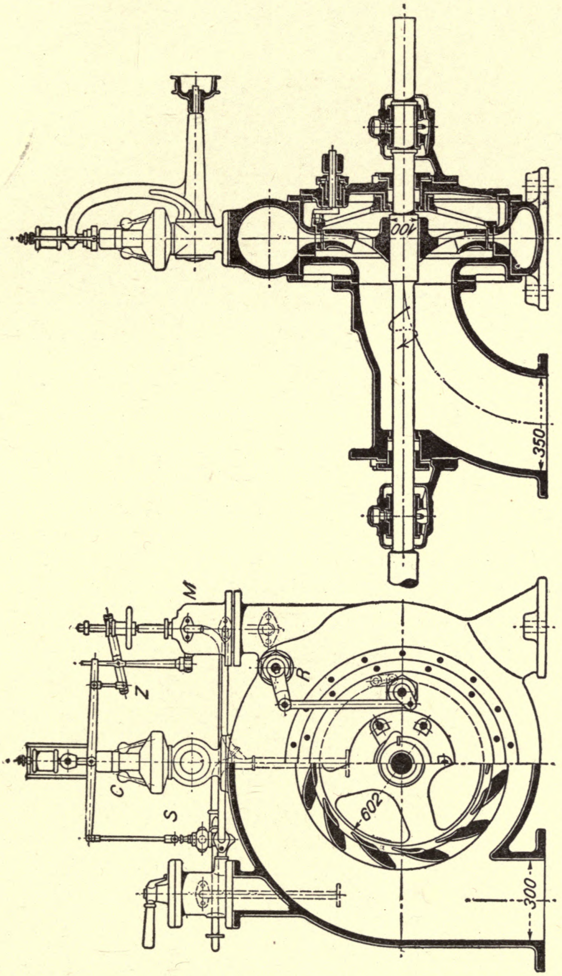


Figs. 26 and 27.—104-H.P. Turbine. Built by the Aktien-Gesellschaft (formerly J. J. Rieter), Winterthur, Switzerland.

Another small single turbine is shown in Figs. 28 and 29. This turbine has also a spiral cast-iron case and is intended to develop 200 H.P. under a head of about 140 ft. and to run at 600 revolutions per minute. The speed is regulated by register gates.

The turbines for the Cataract Power Co., of Hamilton, Ont.,

shown in Figs. 30 and 31, are also single turbines, but in concentric cast-iron cases. The turbines develop 3000 H.P. each under a head of 256 ft., run at 286 revolutions, and give an efficiency



FIGS. 28 and 29.—200-H.P. Turbine. Built by Escher, Wyss & Co., Zurich, Switzerland.

of 80%. To get a small runner diameter and thus a small turbine for the head employed and the required number of revolutions, a low-speed factor was chosen, viz., 0.575. The turbines

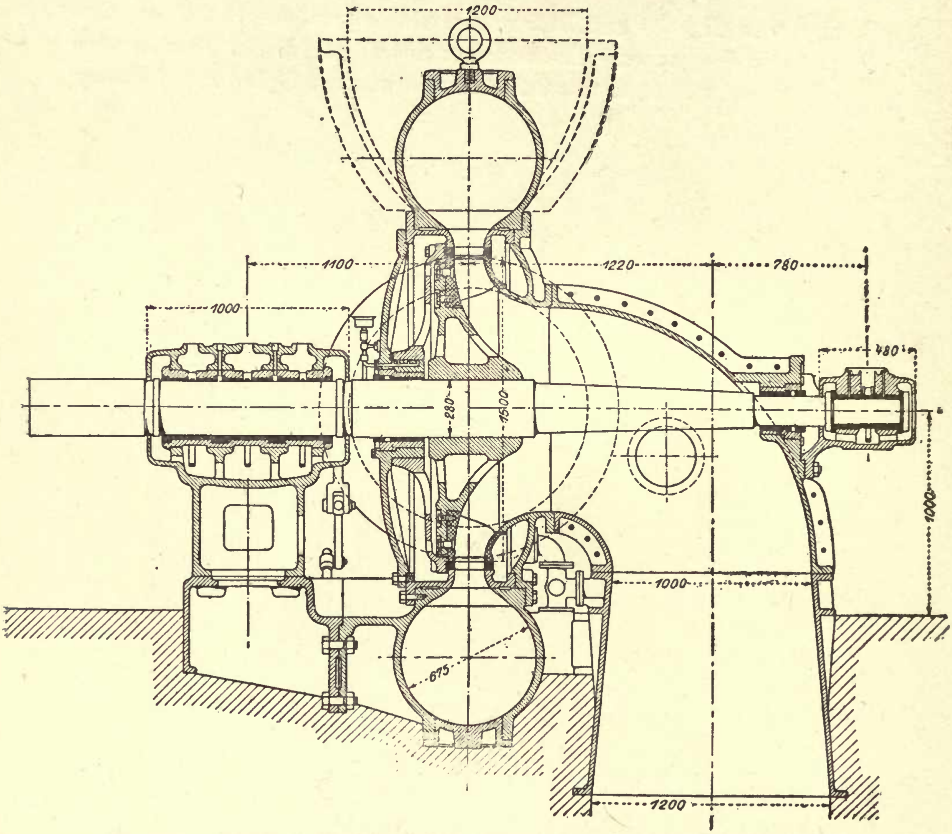


FIG. 30.—3000-H.P. Turbine for the Cataract Power Co., Hamilton, Ont. Built by A. Riva, Monneret & Co., Milan, Italy.

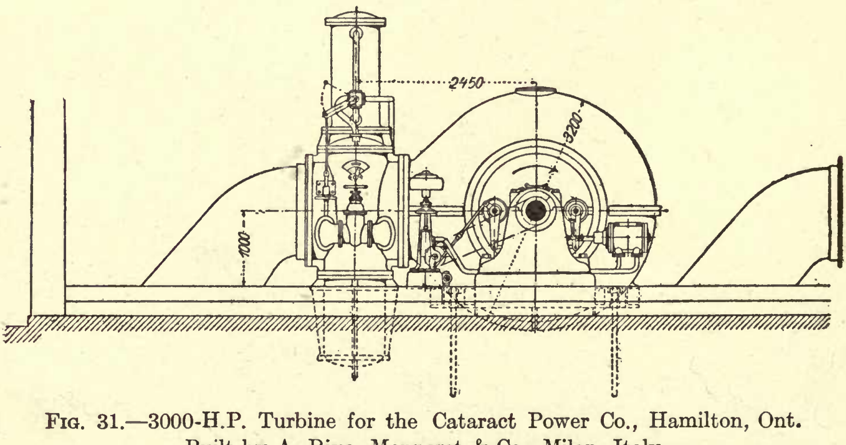
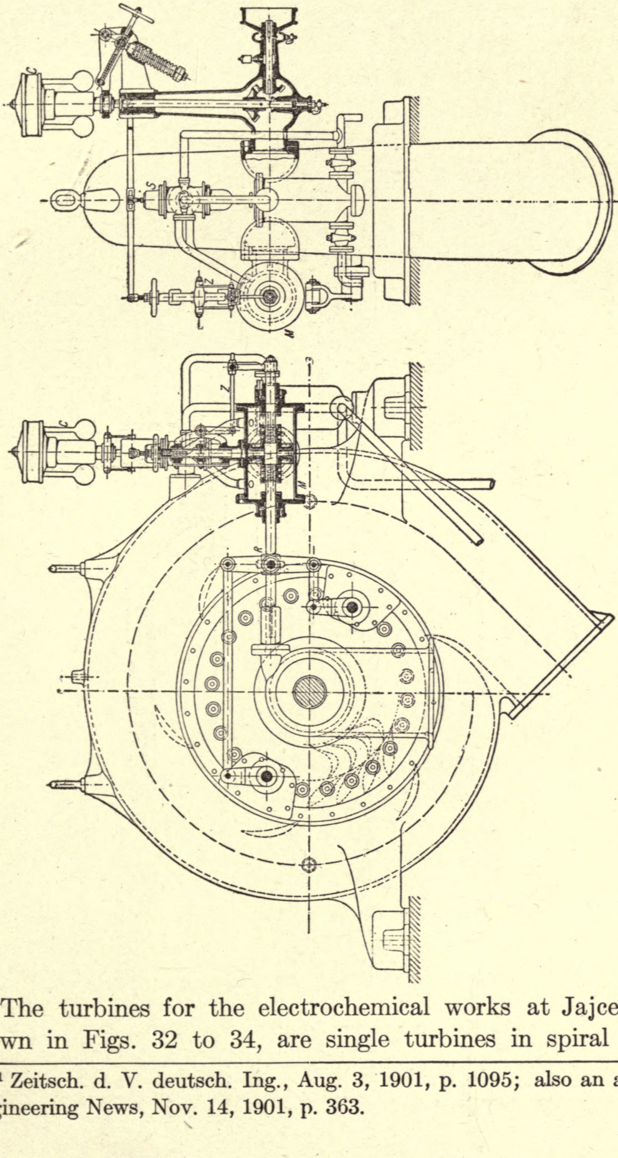


FIG. 31.—3000-H.P. Turbine for the Cataract Power Co., Hamilton, Ont. Built by A. Riva, Monneret & Co., Milan, Italy.

are regulated by register gates of Zodel's design, as shown in Figs. 3 and 4. As will be seen from Fig. 30, the turbines are provided with thrust-chambers.[1]



FIGS. 32 and 33.—1000-H.P. Turbine for the Bosnische Elektrizitaets-Aktien-Gesellschaft, Jajce, Bosnia. Built by Ganz & Co., Budapest, Hungary.

The turbines for the electrochemical works at Jajce, Bosnia, shown in Figs. 32 to 34, are single turbines in spiral cast-iron

[1] Zeitsch. d. V. deutsch. Ing., Aug. 3, 1901, p. 1095; also an abstract in Engineering News, Nov. 14, 1901, p. 363.

cases. The plant contains eight units of 1000 H.P. each and two units of 632 H.P., all working under a head of 245 ft. and running at 300 revolutions. The 1000- and 632-H.P. turbines are identical in every respect, except that the former have a width between runner-crowns of 75 millimeters (3 ins.) and the latter a width of 50 millimeters (2 ins.), measured at the entrance rim. The 1000-H.P. turbines, driving three-phase alternators, gave when tested by means of electrical resistances, and assuming the efficiency of the alternators as 95%, an efficiency of 84%

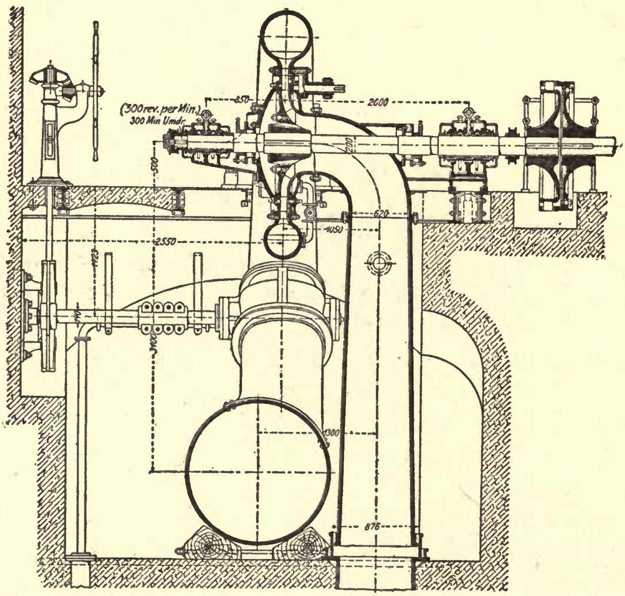


FIG. 34.—1000-H.P. Turbine for the Bosnische Elektrizitaets-Aktien-Gesellschaft, Jajce, Bosnia. Built by Ganz & Co., Budapest, Hungary.

with 0.8 gate-opening. To reduce the runner diameter and with it the whole turbine to a minimum size, for the head and number of revolutions employed, without having to contend with the disadvantages of the action turbine, the low-speed factor of 0.487 was selected; the turbine is therefore a limit turbine. The turbines are regulated by wicket gates of Fink's design and have each four deflector-vanes in the spiral case to throw the water

towards the guide-buckets. These deflectors, being cast solid with the case, also tie the case together and thus help the latter to resist the water-pressure. As will be seen from Fig. 34, the turbines are provided with thrust-chambers. The draft-tube is of cast iron, made in two lengths with a special piece to connect the two lengths at the center, so that the upper length may be removed without disturbing the lower one, which is imbedded in the concrete.[1]

The single turbine in concentric cast-iron case, shown in Fig. 35, differs greatly in its general arrangement from the turbines pre-

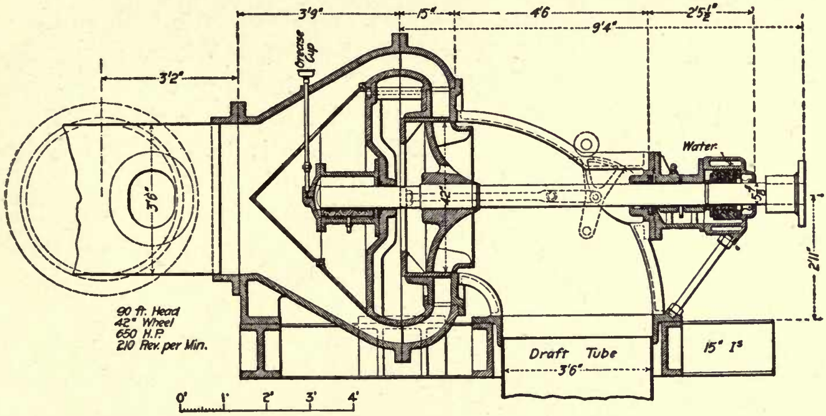


Fig. 35.—650-H.P. Turbine for Messrs. J. & J. Rogers, Ausable Forks, N. Y. Built by Stilwell-Bierce & Smith-Vaile Co., Dayton, O.

viously illustrated. On account of the end or axial direction of the water inlet to the case, the water approaches the guide-buekets in a radial instead of a more or less tangential direction, as is the rule with turbines having a side inlet. The turbine develops 650 H.P. under a head of 90 ft., runs at 210 revolutions, and has a speed factor of 0.5. The regulation is effected by a cylinder

[1] Zeitsch. d. V. deutsch. Ing., Oct. 6, 1900, p. 1354; also Schweiz. Bauz., Feb. 23, 1901, p. 77.

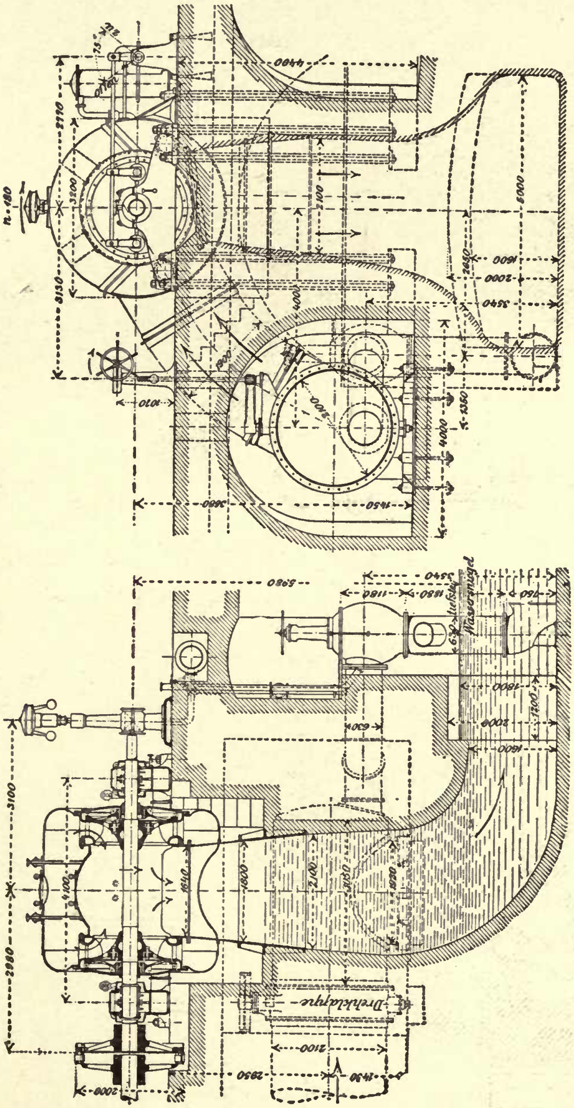
gate and the bearing inside of the inlet is inclosed and lubricated by forcing grease into it. However, there is here no necessity to have one bearing under water, as this can be easily avoided by dividing the end-inlet nozzle into two branches and connecting them to the turbine-case at radially opposite sides.

Double turbines on horizontal shafts may either discharge towards each other or in directions opposite to each other, and the two turbines may be either in a single case common to both or have each a separate case.

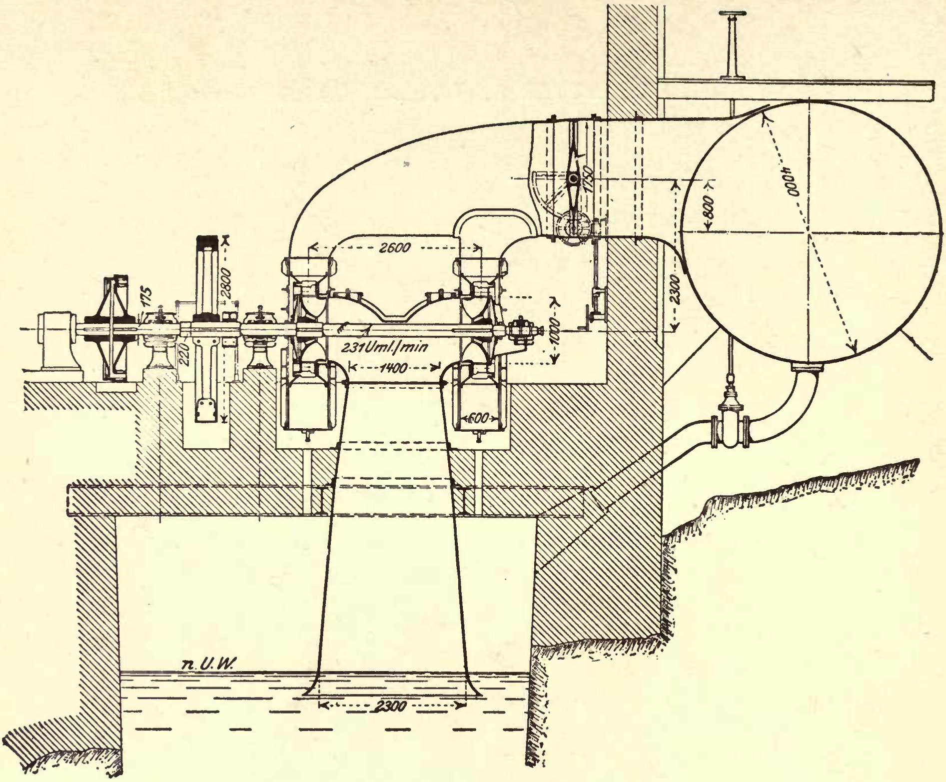
The Paderno power-house of the Edison Electric Co., of Milan, Italy, contains seven turbine units like the one shown in Figs. 36 and 37. These double turbines discharge towards each other, are inclosed in a concentric steel-plate case common to both and each unit develops 2160 H.P. under a head of 94.5 ft. and runs at 180 revolutions. The turbines are of the inflow reaction type, regulated by register gates of Zodel's design, and have shown an efficiency of 82%. Part of each draft-tube is formed by the power-house foundations.

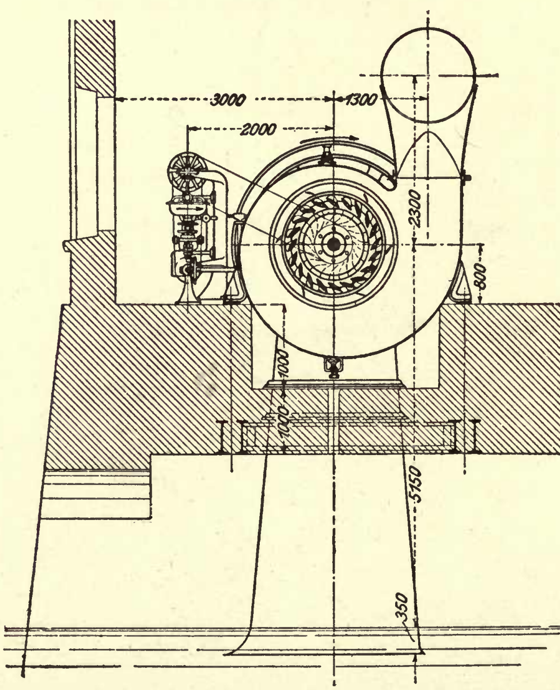
In Figs. 38 and 39 is shown one of the double turbine units, built for the carbide works at Notodden in Norway. The runners discharge towards each other, but each turbine has a separate spiral case made of steel plate. The plant at present contains two such units, each developing 1200 H.P. under 60.7 ft. head and running at 231 revolutions. The speed regulation is effected by wicket gates.

The turbine shown in Figs. 40 and 41 is a double turbine, the runners of which discharge in directions opposite to each other. The runners are placed back to back and have only one guide-ring, common to both runners, and the guide-ring is surrounded by a spiral cast-iron case. This double turbine develops 160 H.P. under 39.4 ft. head and runs at 400 revolutions, but although used in this instance under a low head, the general arrangement is that of a turbine for medium heads. The speed regulation is effected by a peculiar design of gate. As has been stated already, the entrance end of the guide-vanes not only swings like a wicket gate but also moves across the guide-bucket like a register gate, so that the inner end of the movable part of one vane meets the discharge or inner end of the stationary part of

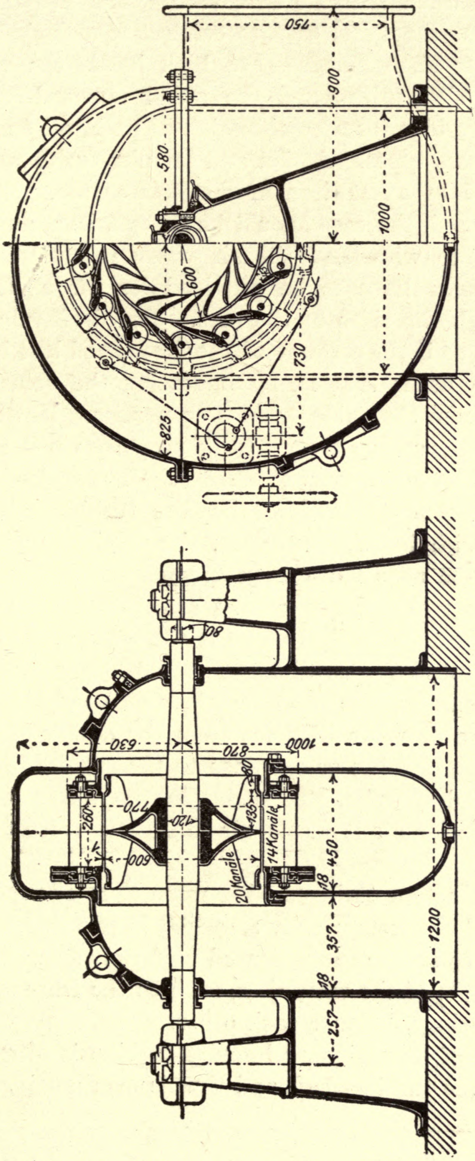


FIGS. 36 and 37.—2160-H.P. Turbine for the Paderno Power-house of the Edison Electric Co., Milan, Italy. Built by A. Riva, Monneret & Co., Milan, Italy.





Figs. 38 and 39.—1200-H.P. Turbine for the Carbide Works at Notodden, Norway. Built by J. M. Voith, Heidenheim, Germany.



FIGS. 40 and 41.—160-H.P. Turbine. Built by Th. Bell & Co., Kriens, Switzerland.

the next vane when the gate is closed.[1] The two draft-elbows are united below the spiral case, to connect to a single draft-tube.

Turbines for High Heads.—For high heads the writer considers the impulse turbine and the radial outward-flow full- or partial-action turbine, or a combination of both, all on horizontal shafts, as the best types. Action and impulse turbines are nearly always inclosed in a cast-iron or steel-plate case. If a draft-tube is to be used, the case should be of cast iron, air-tight, and strong enough to withstand the air-pressure from the outside, and stuffing-boxes have to be provided for the shaft where it projects through the sides of the case. The draft-tube has here the same effect as if the amount of the draft-head had been added to the pressure-head; that is, by increasing the velocity of the water issuing from the guide-buckets or nozzles. However, as an action or impulse turbine cannot work properly when submerged, the water in the turbine-case or draft-tube has to be kept below and clear of the runner and for impulse turbines also clear of the nozzle. With impulse turbines, the vertical distance from the center of the nozzle discharge-opening or the mean vertical distance, where two or more nozzles are used, to the surface of the water in the case or draft-tube is the only part of the head that is really lost, but this distance should rarely exceed 12 ins. In the same manner, the head lost in an action turbine is the vertical distance or mean distance from the center of the guide-bucket discharge opening or openings to the surface of the water in the case of draft-tube. To maintain the water level at the desired elevation an automatic air-admission valve is used, as already stated.

Partial-feed action turbines with more than one guide-bucket have these guide-buckets usually arranged in two sets, spaced 180° apart, or in three or more sets, spaced uniformly around the circle, so that the effects of the water-jets, such as the transverse strains on the turbine-shaft, balance each other.

The runner of an action or impulse turbine is often mounted on the end of the turbine-shaft and thus is overhanging, and when

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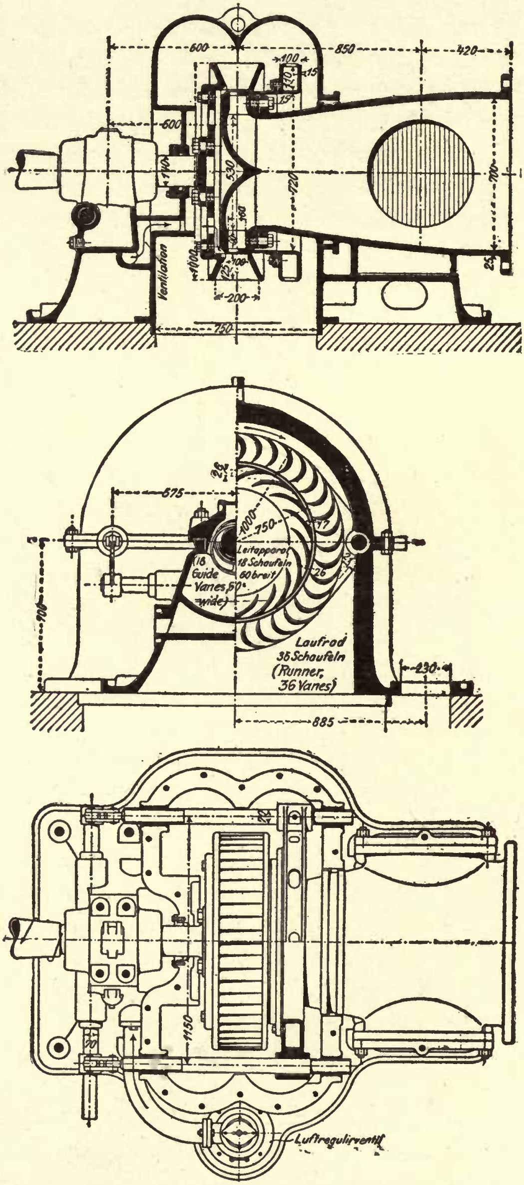
[1] Schweiz. Bauz., April 27, 1901, p. 178, Fig. 61.

driving a dynamo the runner is frequently keyed to the end of an extension of the dynamo shaft.

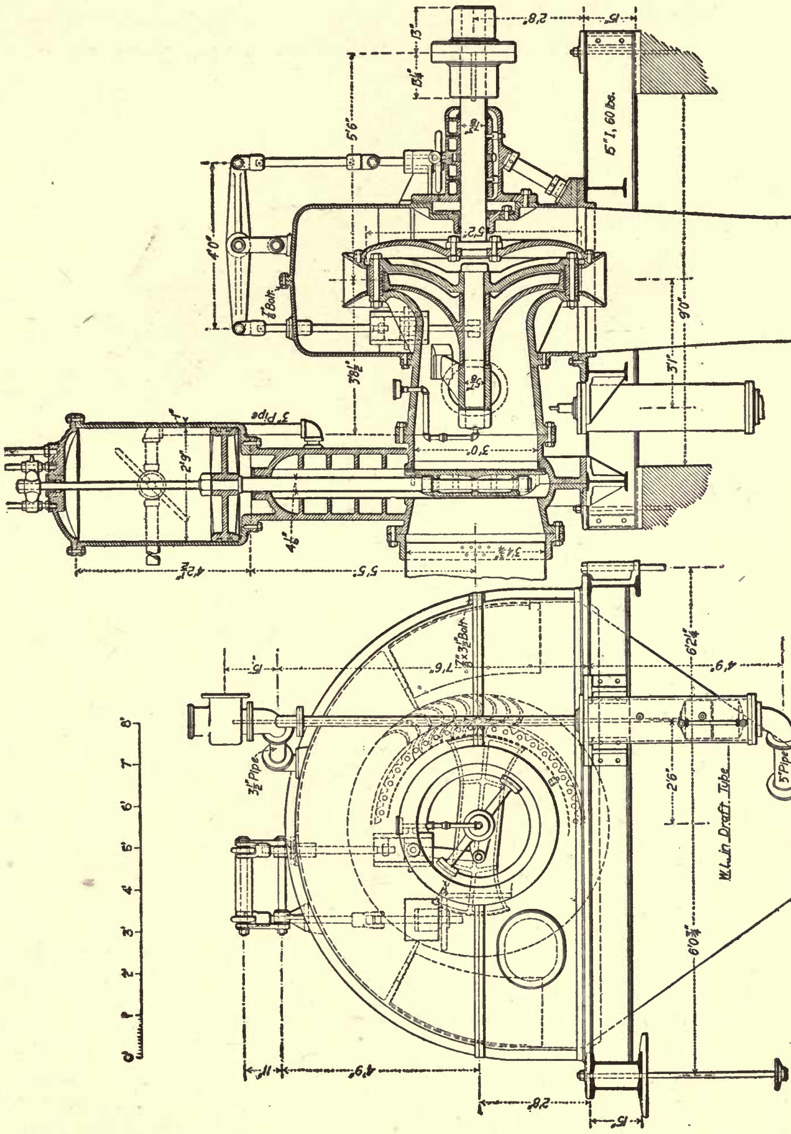
Figs. 42 to 44 show an action turbine built for Elektrizitaetswerk at Schwyz, Switzerland. The plant contains four such turbines, each developing 600 H.P. under a head of 246 ft. and running at 400 revolutions. The runner is overhanging and inclosed in a cast-iron case. The case has on the inside a ridge or cutting edge, so as to divide the water discharged from the runner and divert it sideways, to prevent it from being thrown back on to the runner (Fig. 42). Each turbine is provided with a draft-tube and air-admission valve, the air entering through the channel marked ventilation in Fig. 42. The regulation is effected by a cylinder gate, stiffened by a hollow cast-iron ring.

The turbine shown in Figs. 45 and 46, built for the Ouiatchouan Pulp Co., Ouiatchouan, Que., is a partial-action turbine having two sets of guide-buckets spaced 180° apart. The turbine develops 1000 H.P. under a head of 240 ft. and runs at 225 revolutions. The runner is overhanging and inclosed in a cast-iron case. The turbine is provided with a draft-tube and air-admission valve. The principle of the air-admission valve is very simple and is clearly shown in Figs. 45 and 46. A vertical pipe of about 12 ins. diameter is placed alongside the turbine, with its top end connected to the turbine-case and its bottom end to the draft-tube, in the same manner as a water-gage on a boiler. In this pipe is a copper float, and when the water level in the turbine-case rises this float rises also and opens an air-admission valve on top of the turbine-case, and the water level and float at once drop and the valve closes again. When the turbine is running under a steady load, the valve usually comes to rest in a position where it supplies air at the same rate as it is absorbed and carried off by the water. A gage-glass or water-gage should be attached to the turbine-case or float-chamber so it can always be seen whether the air-valve is maintaining the water at the desired level.

The regulation of this turbine is effected by two slides formed by the ends of a cast-iron beam or girder which swings on a pivot placed in line with the turbine-shaft. By moving this girder, the entrance openings of one or more guide-buckets of each of



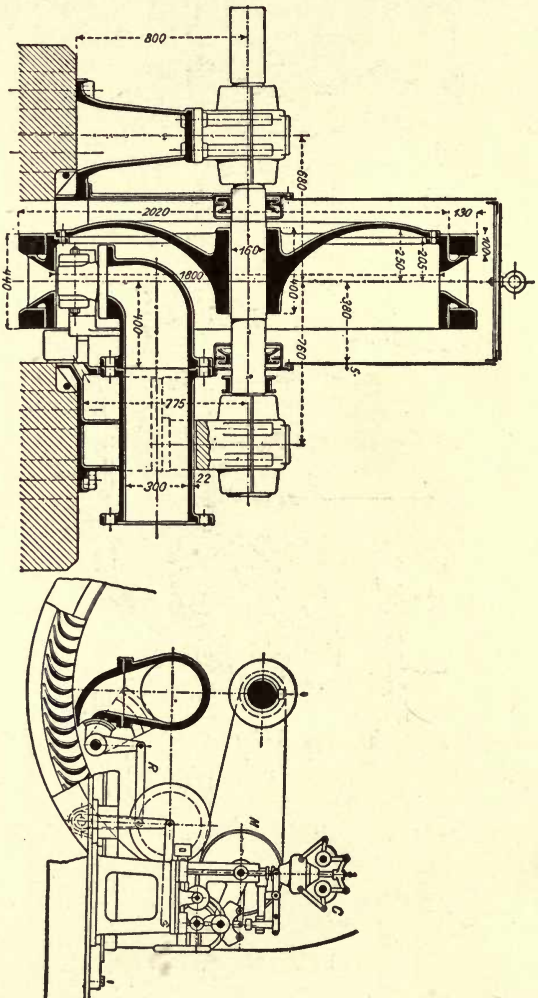
FIGS. 42 to 44.—600-H.P. Turbine of the Elektrizitaetswerk, Schwyz, Switzerland. Built by Th. Bell & Co., Kriens, Switzerland.



FIGS. 45 and 46.—1000-H.P. Turbine for the Ouiatchouan Pulp Co., Ouiatchouan, Que. Built by Stilwell-Bierce & Smith-Vaile Co., Dayton, O.

the two opposite sets of guide-buckets are closed or opened simultaneously by these slides.

Figs. 47 and 48 show a partial-action turbine built for the



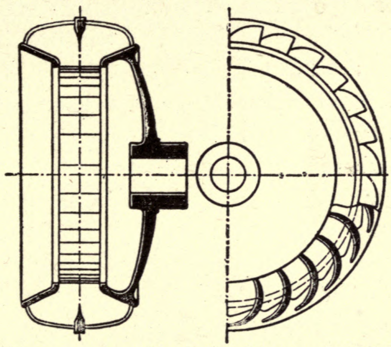
FIGS. 47 and 48.—1000-H.P. Turbine for the Walliser Industrie-Gesellschaft, Vernayaz, Switzerland. Built by Piccard, Pictet & Co., Geneva, Switzerland.

Walliser Industrie-Gesellschaft, Vernayaz, Switzerland. The plant contains six such turbines, each developing 1000 H.P. under a head of 1640 ft. and running at 500 revolutions.

The peripheral speed of the runner is normally 184 ft. per second, but should the governor fail to act, with no load on the turbine, this speed may run up to nearly twice as much. To prevent the runner from bursting, two heavy steel rings are shrunk on to it, which at the same time aid the regulation by acting as fly-wheels.

The lower half of the turbine-case is of cast iron and the upper half of steel plate. The turbine has but one guide-bucket and the regulation is effected by a slide, hinged as to be nearly balanced in all positions.

In Figs. 49 and 50 is shown a runner of a spoon-turbine, a type which may be said to stand between the action and the



FIGS. 49 and 50.—Runner of a Spoon-turbine. Built by Piccard, Pictet & Co., Geneva, Switzerland.

impulse turbine, as, like in an action turbine, the water flows radially outward through the runner, while the runner-buckets have the general shape of the impulse-turbine bucket and, like the latter, are provided with a ridge or cutting edge to divide the water-jet into halves.

For impulse turbines of the Pelton type, the use of draft-tubes gives not only the advantage of added head but also gives a decreased friction of the runner in the surrounding air and a decreased windage; that is, the action of the runner as a centrifugal blower, both due to the decreased density of the air in the case.

With a head of 1000 ft. and 400 revolutions per minute, an impulse turbine would have a diameter of bucket circle of 5 ft.

6 ins. and an outside diameter of about 6 ft., so that an air-tight case could easily be provided for it. With slower speeds or higher heads, requiring larger runners, an air-tight case would be rather expensive, and as the percentage of gain due to the use of a draft-tube is only trifling with heads of over 1000 ft., draft-tubes will hardly be used for impulse turbines larger than about 6 ft. in diameter.

To regulate the speed of an impulse turbine by changing the amount of water supplied to the turbine, American builders employ either a common gate-valve to throttle the water before it reaches the nozzle, which is a very poor arrangement and only used for minor installations, or they employ the needle-nozzle, which is a very simple and efficient device. The needle-nozzle consists of a properly shaped nozzle, inside of which and concentric with it is a movable needle with a cone-shaped end. By pushing this conical end more or less into the exit opening of the nozzle, this opening is more or less reduced. For very high heads or very long penstocks, where a sudden reduction of the size of the nozzle opening would cause dangerous strains in the penstock, a by-pass regulation is employed, the usual arrangement being to have the nozzle connected to the penstock or branch pipe by a flexible joint, and to deflect the nozzle more or less away from the buckets, in proportion to the reduction of the load on the turbine. The flexible pipe-joint of deflecting nozzles, usually a ball joint, used in connection with the high-water pressures considered here, is frequently a source of much trouble. As deflecting nozzles at all times discharge the water required for the maximum load, they are very wasteful when used for variable loads, and to avoid this needles have been employed in some instances in connection with deflecting nozzles. In such cases the needle, usually operated by hand, takes care of the larger fluctuations and the deflection of the nozzle, controlled by the governor, of the smaller fluctuations.

The cross-section of all the nozzles used by American builders is round, while the majority of European builders use rectangular nozzles.

A rectangular needle-nozzle might be employed having a needle to conform to the section of the nozzle, but European

builders always use a tongue in connection with rectangular nozzles.

A tongue-nozzle has two of its sides parallel, while one of the two remaining sides is hinged and forms the tongue, which works like the beak of a bird, as shown in Figs. 51, 52, and 55. As will be seen from the illustrations, both nozzles are provided with a temporary by-pass—that is to say, a by-pass which opens in the same proportion as the nozzle opening is decreased—but the governor slowly closes the by-pass again as soon as the nozzle opening stops to decrease.

The water-pressure, tending to force the tongue outward, is not as great as might be supposed, as at the point where the tongue is located most of the head of the water has already been converted into velocity. The tongue can also be hinged in such a manner as to be partly balanced. For the same cross-sectional area, the perimeter of a rectangular nozzle is greater than the perimeter of a round one, but the presence of the needle in the round nozzle will have the effect that the friction loss in a needle-nozzle will be fully as great as in a tongue-nozzle.

Rectangular nozzles of European builders resemble in shape the turbine guide-buckets and have been made as large as 3×6 ins. at the discharge-opening. In fact the action and impulse turbines are approaching each other more and more in European practice, and will soon be fused into one single type, retaining the best features of both.

To obtain good efficiency, nozzles and runner-buckets must be carefully designed and dimensions and shape adapted to each other.[1]

When draft-tubes are used with by-pass regulation, the discharge from the by-pass must not strike the water surface in the turbine-case or draft-tube, as otherwise it would throw the water into such commotion as to interfere with the proper working of the runner. To prevent this, the by-pass should be located outside the turbine-case, as shown in Fig. 55, or baffle-plates used;

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[1] See paper by Mr. G. J. Henry, Jr., "Tangential Water-wheel Efficiencies," read before the Pacific Coast El. Transm. Assoc., June 16, 1903; also an abstract in Engineering News, Oct. 8, 1903, p. 322.

or spouts so arranged as to carry the by-pass discharge below the surface of the water in the case or draft-tube.

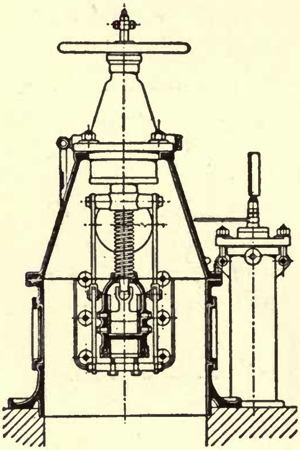

Fig. 51.

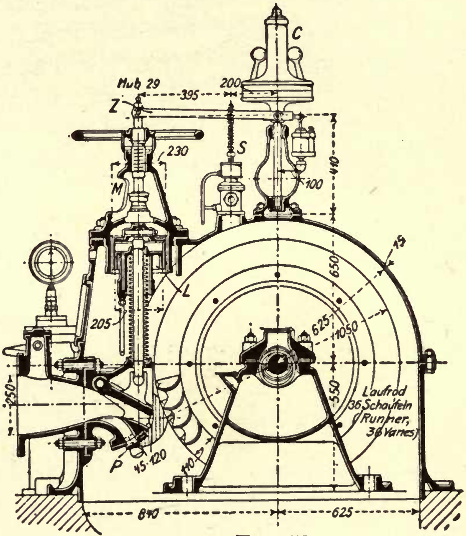

Fig. 52.

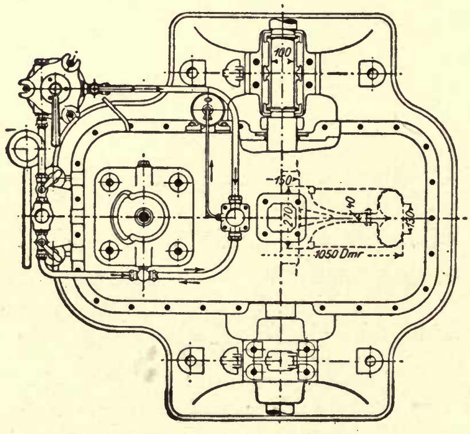

Fig. 53.

Figs. 51 to 53.—360-H.P. Impulse Turbine. Built by Th. Bell & Co., Kriens, Switzerland.

In Figs. 51 to 53 is shown an impulse turbine developing 360 H.P. under a head of 492 ft. and running at 500 revolutions.

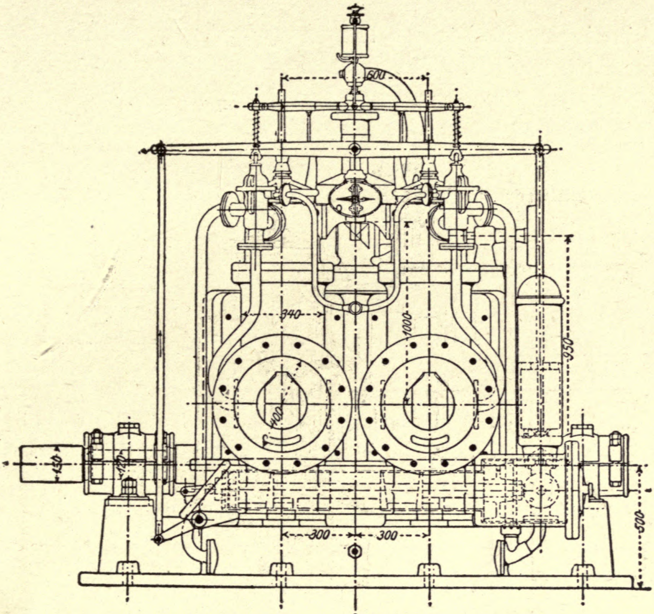

FIG. 54.

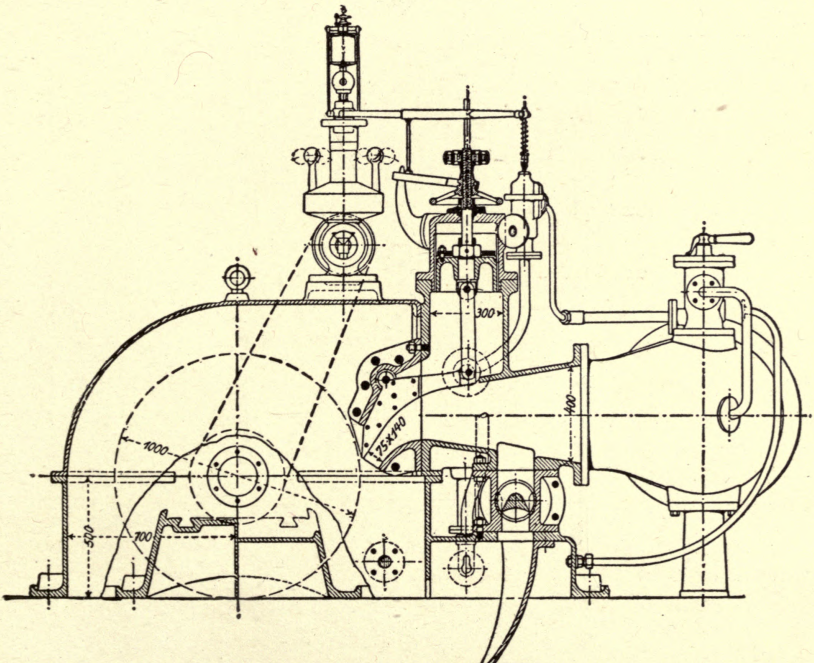

FIG. 55.

The turbine is provided with a temporary by-pass, which is opened and closed by a slide. The operation of the governor will be considered under "Governors and Speed Regulation."

In Figs. 54 to 56 is shown one of the units of the Elektrizitaetswerk Kubel, near St. Gallen, Switzerland. Each unit consists of a double impulse turbine, developing 500 H.P. under a head

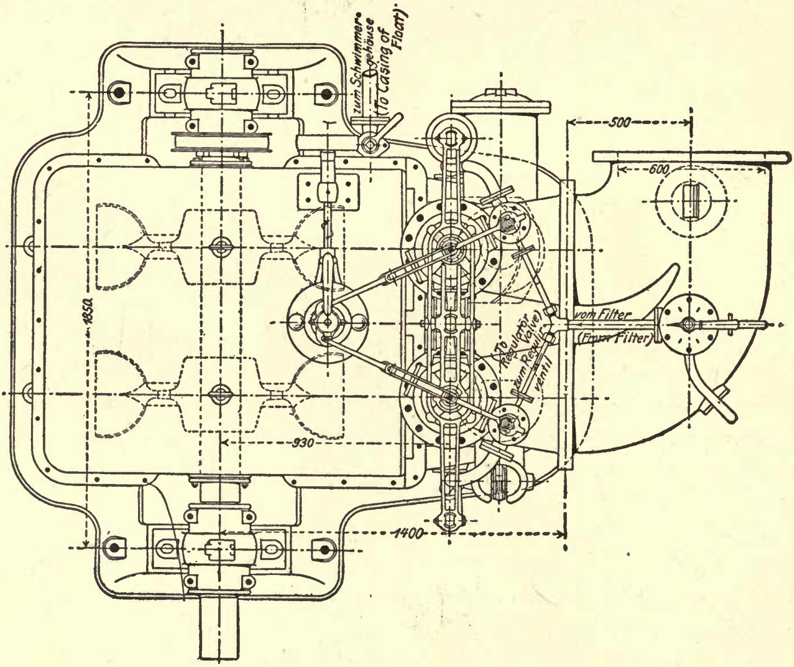

Fig. 56.

Figs. 54 to 56.—500-H.P. Impulse Turbine for the Elektrizitaetswerk Kubel, near St. Gallen, Switzerland. Built by Escher, Wyss & Co., Zurich, Switzerland.

of 295 ft. and running at 375 revolutions. Each unit is inclosed in a cast-iron case and has a draft-tube. The temporary by-pass discharges outside of the case and the water enters the latter sideways and below the water level. The float for operating the air-admission valve is located in a chamber not directly attached to the case. The variation of the water level in the case is less than 2 ins.

One of the most interesting installations of impulse turbines is the electric plant at Vouvry, near Geneva, Switzerland. There are at present four turbines in operation, each developing 500 H.P., running at 1000 revolutions and utilizing a head of 3117 ft., equal to a pressure of 1350 lbs. per sq. in., the highest head of any existing development. Two of the turbines are built by the Société de Constructions Mécaniques of Vevey, Switzerland, and resemble inward-flow action turbines. The two remaining turbines, built by M. Duvillard, Lausanne, Switzerland, resemble impulse turbines of the Pelton type, but the ridge or cutting edge, to divide the water-jet into halves, is placed inside of the nozzle instead of in the runner-buckets, and the jet therefore leaves the nozzle already divided. The outer edge of the very thick cast-iron disk, on each side of which the runner-buckets are located, takes the place of the bucket-ridge. The water flows inward, leaving the buckets at their inner edge. The outside diameter of the runner is $47\frac{1}{4}$ ins.[1]

Manufacture of Turbines.—The manufacture of such modern high-class turbines as have here been illustrated and described would best be taken up by a builder of large, high-grade steam-engines or similar machines, as the work would be of the same character. The large planers, horizontal and vertical boring-mills, etc., of the engine-shop might be used by the turbine department until increased sales warrant the purchase of such tools for the sole use of the turbine-shop. The manufacture of turbine-pumps would naturally form a branch of the turbine-manufacture. Although the guides and runners, as now made by many manufacturers of American turbines, give such high efficiencies under low heads that no appreciable improvement is to be expected in the future, and the guide and runner patterns, involving a great outlay, might be used for modern low-head turbines, yet the writer thinks an engine-builder more likely to produce a high-class turbine than the average turbine manufacturer, as the latter might find it difficult to refrain from using, besides guides and runners, most of the rest of his old patterns and from generally returning to his old shop practice.

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[1] See Engineering News, Nov. 27, 1902, p. 439; also Engineering Record, Nov. 29, 1902, p. 516.

Manufacturers of turbines would best contract for the whole hydraulic equipment of power-plants, building the turbines and governors and subletting penstocks, head-gates, structural-steel work, etc. If this is not done, the manufacturer should furnish the design for the hydraulic equipment outside of the turbines and governors, or be at least consulted in regard to it, as even with turbines of the highest efficiency the total efficiency of a plant may be low, owing to faulty design of the rest of the plant. But outside from the efficiency, the general arrangement of the plant will also influence the accessibility, freedom from interruptions, breakdowns, trouble with floating rubbish, ice, etc. The present practice of some turbine builders to promote water-power companies for the sake of selling turbines and equipping such plants with from two to four times the number of turbines that can possibly be run at times of low water is strongly to be condemned.

The designing of turbines is rather more difficult than the designing of steam-engines, for example. The location of a steam-plant can usually be chosen and the designer can also choose the pressure and the amount of steam and the number of stages of expansions, but the turbine designer has to contend with a location, height of head and amount of water fixed by nature and is required to utilize the head in a single stage. It is therefore important to the turbine manufacturer to have an experienced designing and calculating engineer, occupying a position similar to that of the engineer who makes the calculations and strain-sheets in bridge works.

In going into the manufacture of modern turbines, it would be bad policy to design and make patterns for turbine sizes as orders received may call for. The proper way to bring out a new line of machinery involves a large amount of preliminary office work. To begin with, after collecting all available books, papers, data, etc., on the subject and consulting manufacturers whose machinery is likely to be direct connected to turbines, especially the builders of dynamos, as to speeds, horse-power, etc., a number of different styles of each, the low-, medium-, and high-head turbine, should be designed. Having, after a very careful consideration of all the points involved, selected the style to be

adopted for each range of head, a small-, medium-, and large-size turbine for each range of head should be designed, the dimensions of one or two intermediate sizes calculated, and curves plotted from the results. These curves give at once all the principal dimensions and correct proportions for any size of turbine between the smallest and the largest, and a uniform line of stock sizes may then be chosen for each range of head, say by having the runner diameter in multiples of 3 ins. up to 48 ins. and in multiples of 6 ins. up to 72 ins., all sizes to be made both right and left hand and sizes over 72 ins. to be special and built to order. With such a line of sizes, all conditions of head, power, and revolutions could be met and high efficiencies obtained in all cases.

The turbine-cases should be designed at the start to take guides and runners of different widths of crown or axial dimension and to permit the use of the same patterns of spiral cases for both right- and left-hand turbines. The supporting brackets of the cases should be loose on the patterns, so that the inlet-nozzle may be placed in any desired direction to suit local conditions. Excepting the smaller sizes of turbines, the guide- and runner-vanes with their crowns should be cast in the form rings and bolted to the case and runner-disk or spider respectively. One pattern for each diameter of guide- and runner-ring can usually be made to serve for any width of crown, from the smallest that is permissible to the largest that the case will take and for both right- and left-hand turbines.

The power of the turbine will, of course, vary in direct proportion to the width of the crown, but should the least permissible width of crown still give a power in excess of the required amount the quantity of the water flowing through the turbine may be still further reduced by decreasing the number of guide-buckets, that is, changing the turbine to a partial-feed turbine. However, as reaction turbines do not give good efficiencies with partial feed, action turbines should be employed in such cases.

The width and curvature of the guide- and runner-vanes should be determined for each individual turbine, to suit the quantity of water and height of head to be utilized. It may be repeated here that reaction turbines can be designed to give their maximum efficiency at any desired gate-opening.

The use of cast guide- and runner-vanes requires the making of a separate core-box for each width and shape of vane. Except for limit turbines, where thick vanes are required, the use of steel-plate vanes for both guide- and runner-buckets is to be preferred, and such vanes are used by most European manufacturers. The impact of the water striking the initial edge of the runner-vanes not only causes a loss, but also gives rise to irregularities in the flow of the water, for which reasons this edge should be as thin as possible, a condition favorable to the use of steel-plate vanes. Steel-plate vanes are strong, hard, and smooth, but corrode and pit more readily than cast-iron vanes. Such steel-plate vanes, after being cut to size, are brought to the required curvature by being heated and pressed in cast-iron dies and are then, while still hot, placed in the runner or guide-ring mold and cast in.

Runners and guide-rings are molded by means of molding-machines to avoid deviation from the exact spacing, position and direction of the vanes, and to insure the correct shape of the ring.

To obtain good efficiencies with any type of turbine requires high-class castings, especially the inner surfaces of the bucket- and other water-passages should be perfectly smooth. The material for the guide- and runner-rings should be a hard quality of cast iron for low heads and bronze or steel casting for medium and high heads. The clearance of the runners of reaction turbines should not be larger than $\frac{1}{32}$ in. for small and medium size and $\frac{1}{16}$ in. for the largest size of runners. Action turbines may have larger clearances.

With the American or low-head type of turbine the required power at a given number of revolutions is at present obtained by using one or more turbines on the same shaft, or by varying the width between crowns or axial dimension of the buckets to suit, or, if the necessary reduction of bucket area is only slight, by means of the regulating-gates, and it is likely that these means will be employed in the future for this type of turbines. The runners and guides are usually cast solid, using patterns and core-boxes, although some manufacturers use built-up runners, the vanes being cast singly or in groups of two to three. Vanes

of steel plate, both for guides and runners, are rarely used in American practice.

With the medium head or European type of turbine the required power at a given number of revolutions is obtained by using one or more turbines on the same shaft or by varying the width between the crowns of guides and runners.

With the high-head type—that is, action and impulse turbines—the required power at a given number of revolutions is obtained by using one or more turbines on the same shaft, by using one or more guide-buckets or nozzles for each turbine, and by varying the size of the buckets and nozzles.

CHAPTER VI.

ACCESSORIES TO TURBINES.

The Draft-tube.—Although draft-tubes have only come into general use in comparatively recent years, their great advantage is now universally recognized, as only the draft-tubes made it practically possible to employ turbines on horizontal shafts or to set turbines above the tailwater without losing part of the head.

In considering the effect of the draft-tube it should be borne in mind that:

A given static head may exist either wholly in the form of pressure or pressure-head, or wholly in the form of vacuum or suction or draft-head, or part of the head may be in the form of pressure-head and the remainder in the form of draft-head.

Also, a given head may exist wholly in the form of pressure or static head, or wholly in the form of velocity, or part of the head may be in the form of pressure and the remainder in the form of velocity, and pressure can be converted into velocity or velocity into pressure.

The effect of a draft-tube on a turbine may be compared to the effect of a condenser on a steam-engine, as, like the condenser, the draft-tube removes part of the back or counter pressure due to the pressure or weight of the atmosphere, while the water-column above the turbine acts like the live steam behind the engine-piston. The means employed for partially removing the back pressure are in the case of the turbine the suction caused by a hanging water-column, and in the case of the steam-engine the condensation of steam, creating a partial vacuum.

As the hanging water-column in the draft-tube is counterbalanced or held in equilibrium by the pressure of the atmosphere,

its height or head cannot be greater than that of a water-column exerting on its base a pressure equal to the atmospheric pressure.

At sea level the pressure of the atmosphere is 14.72 lbs. per square inch, or 2119 lbs. per square ft., and this will hold in equilibrium a column of water 34 ft. in height if the water is at rest, but if the water is in motion then the atmospheric pressure has also to counterbalance that height or head, which is contained in the water in the form of velocity, and the height of the column thus balanced will be 34 ft., less the velocity-head, corresponding to the speed of the water, or $34-\frac{c_f^2}{2g}$, in which c_f is the draft-tube speed, that is, the velocity of the water at exit from the lower end of the draft-tube in feet per second and g is the acceleration of gravity, equal to 32.16. With the speed c_f equal to 46.8 ft., the velocity-head becomes 34 ft. and the height of the water-column that will be balanced by the atmospheric pressure becomes $34-\frac{46.8^2}{2\times 32.16}$, which is equal to zero.

In a draft-tube the water-column can, therefore, not have a greater height in feet than $34-\frac{c_f^2}{2g}$, and if the vertical length of the draft-tube is more than that, the water surface in it will remain at an elevation of $34-\frac{c_f^2}{2g}$, and the part of the draft-tube above this level will contain a vacuum, which means a loss of head equal to the height of the empty space.

The considerations and figures just given are theoretical only, and have to be modified in practice.

However, not the whole of this draft-head can be utilized in the turbine, and to obtain the effective draft-head the sum of the various losses has to be deducted from the total draft-head of 34 ft. These losses are due to the friction of the water in the draft-tube, to the change in the speed of the water between the turbine-runner and the lower end of the draft-tube, and to the momentum or velocity still contained in the water while leaving the draft-tube.

It should be mentioned here that the laws governing the action of draft-tubes also apply to the suction of pumps and in this connection may be stated thus: The vertical distance in feet to which

water can be raised by suction is theoretically $34-\frac{s^2}{2g}$, in which s is the velocity of the water with which it enters the suction-pipe in feet per second. From this theoretical height has to be deducted the entrance-head for the suction-pipe and the losses due to friction and bends in the suction-pipe. The result has to be further reduced for the reason that a pump cannot produce a perfect vacuum. It will be seen that a larger suction-pipe with flaring end not only means a decreased friction loss, but also a decreased loss in velocity-head and entrance-head.

Air-bubbles rise in water with a speed of about 12 ins. per second, and to prevent such bubbles from rising inside of a draft-tube and to carry off the air, when starting a turbine with the draft-tube filled with air, the speed per second of the water while leaving the draft-tube should not be less than 2 to 3 ft. As turbines are often required to run continuously with part gate-opening it may be stated that:

The minimum draft-tube speed c_f—that is, the speed with which the water issues from the lower end of the draft-tube—should never be less than 2 to 3 ft. per second, with the minimum gate-opening at which the turbine may be required to run for any length of time or continuously. In practice it is usual to employ a draft-tube speed of 2 ft. for low heads, increasing it to 3 ft. for heads of about 100 ft. and to 4 to 6 ft. for heads of about 500 ft.

The absolute velocity in feet per second with which the water issues from the runner-buckets—that is, the velocity relative to a stationary object—may be taken as

$$c_a=0.285\sqrt{2gH}$$

for large-size turbines and low heads, say 10 ft.; as

$$c_a=0.200\sqrt{2gH}$$

for medium-size turbines and medium heads, say 100 ft.; and as

$$c_a=0.167\sqrt{2gH}$$

for small-size turbines and high heads, say 500 ft., in which H is the total effective head in feet acting on the turbines. This is for turbines of the European type. The writer could find no reliable data in regard to the absolute discharge velocity of American turbines, but it may be stated that these velocities are higher for the American than for the European type.

The heads corresponding to these discharge velocities are entirely lost, except when draft-tubes are used, which discharge the water at lower speeds; or, stating this in another way, of two similar turbines working under the same head, but one set above the tailwater, and provided with a properly designed draft-tube, while the other is set on or below the level of the tailwater, and not provided with a draft-tube, the turbine using the draft-tube will always give the higher efficiency, and the efficiency of a turbine working on or below the level of the tailwater may be increased by the addition of a draft-tube, as the draft-tube acts in exactly the same manner as the Boyden diffuser, making power available out of the water discharged from the runner by retarding the velocity of the water.

A few examples will illustrate this. Assuming the heads H to be 10, 100, and 500 ft. and the corresponding draft-tube speeds c_f, equal to 2, 3, and 4 ft., then the velocity-heads will be

$$h_f=\frac{c_f^2}{2g}, \text{ or } 0.062;\ 0.140 \text{ and } 0.248 \text{ ft. respectively.}$$

If h_a is the velocity-head for the absolute velocity of discharge from the runner-buckets c_a, and G is the gain in head, due to the retardation of the water by the use of a draft-tube, then we have:

$H=$	10 ft.	100 ft.	500 ft.
$c_a=$	$0.285\sqrt{2gH}$ $=7.23$ ft.	$=0.200\sqrt{2gH}$ $=16.05$ ft.	$=0.167\sqrt{2gH}$ $=29.97$ ft.
$h_a=\frac{c_a^2}{2g}=$	$0.285^2\times H$ $=0.796$ ft.	$=0.200^2\times H$ $=4$ ft	$=0.167^2\times H$ $=13.94$ ft.
$G=\frac{c_a^2-c_f^2}{2g}=h_a-h_f=$ ...	0.734 ft.	$=$ 3.86 ft.	$=$ 13.69 ft.
G in per cent of $H=$	7.34%	$=$ 3 86%	$=$ 2.74%

The gains here shown are theoretical only and cannot be reached in practice, but a well-arranged draft-tube should realize 75% or more of the theoretical gain.

Conical or flaring draft-tubes, changing the speed of the water gradually, should always be used, and the present practice of having the same cross-sectional area for the entire length of the draft-tube must be condemned, as the water, issuing from the runner with a velocity several times greater than the velocity of the water-column in the draft-tube, strikes the latter and is so suddenly retarded that most of the power made available by the retardation of the water is lost in the shock.

It must also be borne in mind that to reduce the speed with which the water leaves the runner to the speed with which it leaves the draft-tube requires a certain minimum length of draft-tube, this minimum length being the greater the larger the diameter of the draft-tube is.

No reliable data seem to have been published regarding the minimum length of draft-tube required for a given reduction in the velocity of the water, and the writer has therefore compiled the following table for round draft-tubes of different diameters. The table gives the greatest permissible angle of flare, that is, the angle which the opposite sides or walls of the draft-tube form with each other, also the approximate increase in diameter and area in a length of 10 ft.

MAXIMUM FLARE OF DRAFT-TUBES.

Small End.		Angle of Flare.	Large End.		Proportion of Areas.
Diameter.	Area.		Diameter.	Area.	
2 ft	3.14	15°	4.5 ft.	15.90	1:5.05
4 "	12.57	23°	8.0 "	50.26	1:4.00
6 "	28.27	29°	11.0 "	95.03	1:3.36
8 "	50.26	33°	14.0 "	153.94	1:3.06
10 "	78.54	36°	16.5 "	213.82	1:2.72
12 "	113.10	38°	19.0 "	283.53	1:2.51

In general, it may be said that a short draft-tube may have a greater angle of flare than a long draft-tube.

It will sometimes prove an advantage to increase the length of the draft-tube, without increasing the draft-head, by curving or inclining the draft-tube, as this not only gives a greater length in which to reduce the velocity of the water, but also permits

to discharge the water in the desired direction, besides an inclined draft-tube may often save some of the tailrace excavation. On the other hand, the water in an inclined draft-tube will crowd towards the lower or bottom side of the tube, and to keep the tube running full, the speed of the water has to be higher than in a vertical draft-tube, and this increase in speed must be the greater the more the direction of the centre line of the draft-tube is off the vertical.

The draft-head that can be empolyed in practice is only a part of the theoretical limit of 34 ft. less the velocity-head, the practical limit decreasing with increasing diameters of draft-tube.

The writer, by computing the equivalents in English measurement of the metric values of a table given in a German standard work on turbines and plotting a curve from the results, has deduced the following table of the greatest draft-head that may be employed in practice, under average conditions, and for different diameters of draft-tube.[1]

Diameter of draft-tube in ft..	0.5	1	1.5	2	2.5	3	3.5
Draft-head in ft	31	29.5	28.1	26.7	25.3	23.9	22.6

Diameter of draft-tube in ft	4	4.5	5	6	7	8
Draft-head in ft	21.4	20.2	19	17	15.4	14.2

Diameter of draft-tube in ft.......	9	10	11	12	13	14
Draft-head in ft.	13.2	12.3	11.4	10.6	9.8	9

From the draft-heads here given the velocity-head, corresponding to the speed with which the water leaves the draft-tube, or $h_f=\frac{c_f^2}{2g}$, has to be deducted, and the result will be the greatest permissible draft-head to be employed. For conical draft-tubes this table shows the greatest height above the tailwater at which a given diameter of draft-tube may be used; this height is, of course, to be corrected for the velocity with which the water is

[1] See Meissner. Hydraulische Motoren, vol. 2, p. 212.

discharged from the lower end of the draft-tube in the same way as just stated. For example, a conical draft-tube must not be more than 9 ft. in diameter at a height of $13.2-\frac{c_f^2}{2g}$ ft. above the tailwater level.

Under average conditions these figures must not be exceeded if perfect working of the draft-tube is to be insured, but in general a higher draft-head may be used with turbines with steady loads and always working with their full capacity, while with turbines liable to great and violent fluctuations in their loads and working at times only with small fractions of their full capacity the draft-head should be smaller than given in the table.

A draft-tube will work better in connection with a vertical turbine discharging directly into the tube than with a horizontal turbine connected with the tube by a draft tee or elbow. Some engineers therefore use for horizontal turbines only from 66 to 75% of the permissible draft-head given in the table, but this reduction is not necessary.

Where it is essential, a higher draft-head may be employed by using two smaller draft-tubes instead of one larger one, or by having the upper part of the draft-tube cylindrical and of small diameter and beginning the conical part at a point near enough to the tailwater level to insure the proper action of the draft-tube.

When a draft-tube is used in connection with a turbine set in an open turbine-chamber, then the height of the draft-head is limited by the depth of the water above the turbine. As has already been stated under "Turbines for Low Heads," a certain minimum depth of water above the guide-buckets is required to prevent the air from being sucked through the turbine. It will readily be seen that the use of a draft-tube not only increases the speed with which the water enters the guide-buckets, but that the tendency of the water to form funnels and of the air to be sucked through the turbine is very much increased, and to offset this tendency it is necessary to have a proportionally greater depth of water above the guide-buckets. It may therefore be stated that a turbine set in an open chamber and using a draft-tube should have a depth of water above the highest point of

the discharge rim of the guide-bucket ring at least equal to one half the total effective head (H) acting on the turbine.[1]

When a very high draft-head is employed, especially in connection with large draft-tubes, the water has a tendency to pulsate or oscillate in the draft-tube, and if the turbine is subject to sudden changes in load and provided with a quick-acting governor, the pulsations will increase to such an extent that they not only have the effect of a varying draft-head acting on the turbine, and thus be detrimental to good speed regulation, but these pulsations may even wreck the turbine. With the turbine-gate closing quickly, the momentum of the water-column in the draft-tube will cause that column to break, creating a vacuum, but as soon as the momentum has been arrested, the atmospheric pressure will throw the water-column upward again, striking the turbine-runner with great force.

It should be stated here that conical draft-tubes are less liable to pulsations, and can therefore be employed with greater draft-heads, than cylindrical draft-tubes. Conical draft-tubes are also better able than cylindrical ones to expel the air and form the draft when the turbines are started and to retain the draft when the turbines are running with light loads.

In a large and long draft-tube, when the turbine is started with the draft-tube filled with air, or when running with only a small fraction of the full load, the water is liable to crowd to one side and tumble or drop through the draft-tube without expelling the air or forming a suction, and the draft-head is thus lost. Curved or inclined draft-tubes are more likely to act in this manner than vertical draft-tubes. It is very probable that many turbines may be found running with their draft-tubes partly filled with air, and thus losing the entire draft-head without the fact being known to the parties in charge of the plant.

Gates for closing the lower end of the draft-tube and used for filling them with water before starting the turbines, or for reducing the discharge area when working with part load, have been used to some extent in Europe, but they are expensive and ponderous for large turbines, and it is not probable that they will be adopted in America.

[1] Mueller. Francis-Turbinen, p. 12.

The dip, or the distance to which a vertical draft-tube reaches below the level of the tail-water, should be 6 ins. to 12 ins. for short and small draft-tubes, increasing to 20 ins. to 24 ins. for long and large draft-tubes. In general, it may be said that a greater dip permits a greater draft-head.

The object of the dip is, of course, to seal the draft-tube against the inrush of air. This water seal is of importance, especially while the turbine is started and the draft formed, and it is therefore essential that the draft-tube reaches below the water level at all stages of the tailwater.

To facilitate the escape of the water from a vertical draft-tube, it is advisable to curve the lower end of the walls of the tube outward in a trumpet shape, as shown in Fig. 38. An inclined or curved draft-tube having a discharge in a direction considerably off the vertical, or in a horizontal direction, should be especially protected against the inrush of air by having a scoop-shaped projection at its discharge end, formed by extending the upper side or wall of the draft-tube, as shown in Fig. 24.

The draft-tube should be strong enough to withstand the atmospheric pressure to which it is subjected, and which tends to collapse the draft-tube. The pressure, p, in pounds per square inch, at any vertical distance, d, in feet, above the tailwater is equal to $p=\frac{14.72}{34}\times d$, but can, of course, never exceed the pressure of the atmosphere, or 14.72 lbs.

Draft-tubes are usually made of steel plate, with lap-riveted telescoping courses, and must be thoroughly air-tight, as any leakage of air into the draft-tube will destroy the vacuum and cause the loss of the draft-head. Draft-tubes are nearly always round in cross-section, as this form is the most convenient to manufacture and to make tight, and offers the greatest resistance against collapse. Where a concrete substructure is used for the power-house, it will often be found economical both in first cost and maintenance, to dispense with a metal tube altogether and to mold the draft-tube directly in the concrete of the substructure.

Stop-valves for Turbines.—When it is desired to shut down or stop a turbine unit without the use of the head-gate it is the common practice to close the regulating-gates. However, this

method will not permit the turbine to be taken apart or repaired, and in connection with high heads the regulating-gates are rarely tight enough to bring the turbine to a standstill. It is therefore advisable to have a separate stop-valve in the supply-pipe near the turbine, by which means the water may be quickly shut off and the turbine stopped, even if some accident or the accumulation of ice make the closing of the head-gate impossible. Where more than one turbine unit is connected to the same penstock, such stop-valves are essential, as otherwise the necessity of repairing one unit would require the stopping of all the units connected to the same penstock.

The opening and closing of such stop-valves should be sufficiently slow to permit the water to change its velocity without greatly decreasing or increasing the pressure in the penstock.

In America the stop-valves employed for turbines are nearly always gate-valves. The smaller sizes of such valves, especially when used in connection with low heads, are, as a rule, provided with a screw-spindle and operated by hand. Large gate-valves, especially when used in connection with high heads, require so great a power to operate them quickly that it becomes necessary to have some kind of motor for this purpose capable of furnishing a large power for a short space of time.

In some water-power electric plants an electric motor is used, geared to the screw-spindle of the valve. This, however, requires a very large motor and involves either the installation of a storage-battery or the use of current from some outside source to be able to start a unit when the whole plant is shut down.

More frequently used is a hydraulic lift mounted above the gate-valve and having its piston-rod coupled directly to the valve-spindle, as shown in Figs. 31 and 46. Where the head employed is high enough, the pressure water may be taken directly from the penstock, but for lower heads a pressure-pump and weighted accumulator are required, and in this case oil or a mixture of glycerine and alcohol is frequently used as a pressure fluid instead of water. An accumulator consisting of a closed tank or receiver partly filled with water, which is subjected to air-pressure, is not to be recommended for this purpose.

The use of air instead of a liquid in the operating cylinder

of a gate-valve is not advisable, as owing to differences in resistance encountered at different parts of the valve travel, the valve may alternately stick and jump forward, and is thus likely to produce serious shocks in the penstock, due to the sudden changes in the clear passage area of the valve. For the same reason a back pressure should always be maintained against the lower side of the piston when the gate moves downward, to prevent it from dropping by its own weight.

A by-pass valve is often employed in connection with a gate-valve, but even if of proportionately large size it can only partly balance the pressure against the main-valve disk, owing to the leakage through the regulating-gates.

The gate-valve has the advantage to present a clear passage area to the flow of the water, and if in good condition, to be perfectly tight, but it requires a great amount of power to operate it and its first cost is very high.

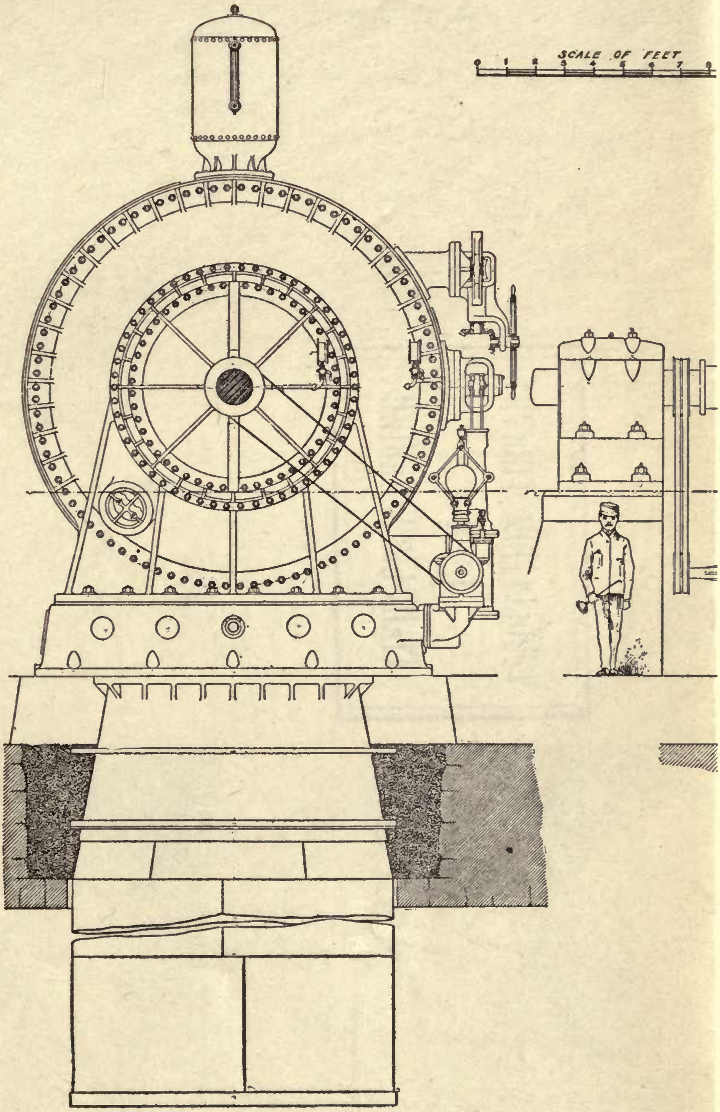
In Europe the stop-valves employed are nearly always wing-gates, operated by hand by means of a worm-gear and worm, as shown in Figs. 22 and 23 and 36 to 38.

The wing-gate is practically balanced in all positions, thus requiring little power to operate it, and its first cost is low; but the passage area for the water is always obstructed by the wing and it is impracticable to make it perfectly tight. No tests seem to have been made to ascertain what amount of resistance the wing offers to the flow of the water.

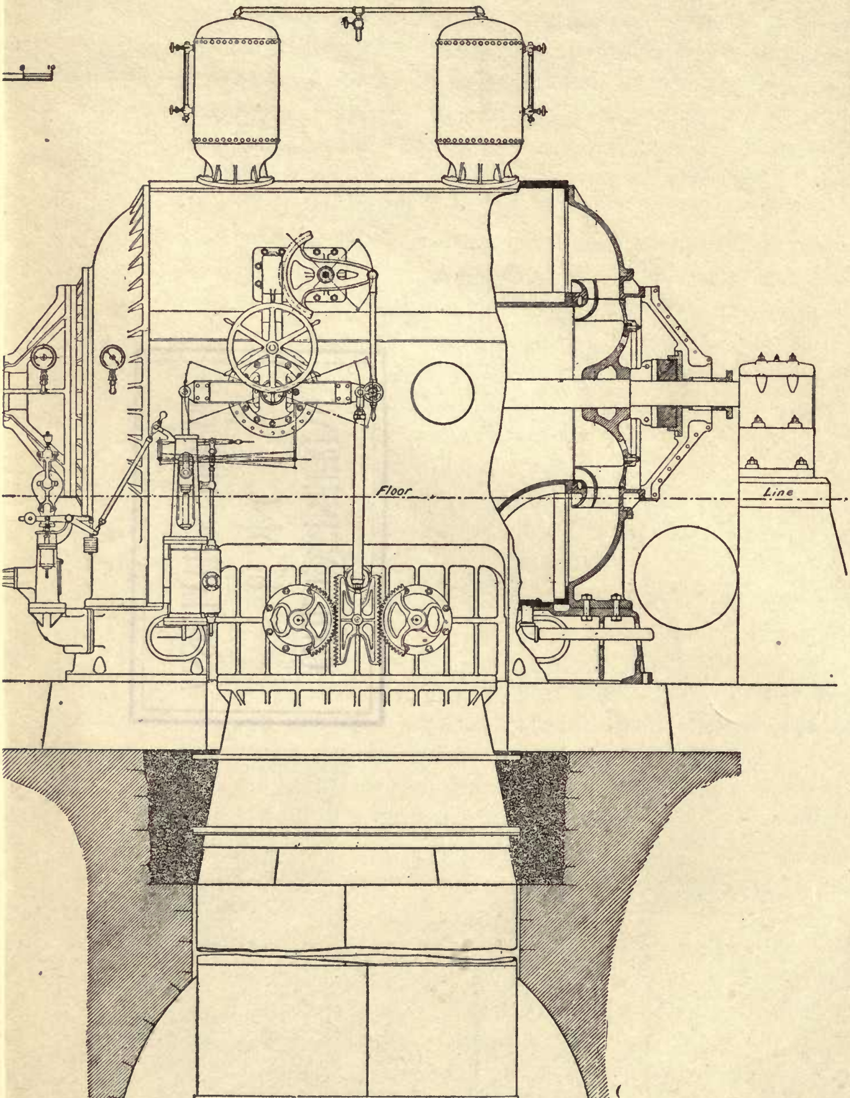
In a few instances, where the turbine units are connected to the penstock by elbows of short radius, angle stop-valves have been employed, the water passage area being the annular space between the outside of the valve-disk and the inside of the valve-body.

In cold weather the shutting down of a plant over night or over Sunday should be effected by means of the head-gates and the penstock emptied of water, as the water when stationary might freeze solid and destroy the penstock.

Throttling-gates for Speed Regulation.—To regulate the speed of turbines by means of throttling the water, either in the penstock above or in the draft-tube below the turbine, is, next to the by-pass regulation, the most wasteful arrangement possible, as has already



FIGS. 57 and 58.—6000-H.P. Turbine for the Shawinigan Water and I
diam. of runner 6 ft 4 in.; diam. of penstock, 11 ft.; cubic



ower Co., Shawinigan Falls, Que. (Height of head, 125 ft.; rev. per min., 180; feet of water per sec., 576; vanes in guide-ring, 36; vanes in runner, 32.

[To face page 135

been stated in connection with the European practice, and this method should never be employed.

The large turbine units installed in the main power-house of the Shawinigan Water and Power Co., at Shawinigan Falls, Que., and shown in Figs. 57 and 58, are regulated in this manner, having two wing- or throttling-gates inside of the discharge-opening of the draft-tee or at the upper end of the draft-tube, and as the plant is of considerable importance and the turbines are some of the most powerful that have been built up to the present time, it would be well here to consider the matter at some length.

According to Mr. W. C. Johnson, chief engineer of the Shawinigan Water and Power Co., each unit consists of a pair of radial inward-flow reaction turbines on horizontal shaft, and develops 6000 effective H.P., under 125 ft. head, and at 180 revolutions per minute, using 576 cu. ft. of water per second, and therefore having an efficiency of 73.6%, when working with full capacity; but this figure is very low for such a turbine, and an efficiency of about 78% will probably be obtained in practice. Taking here a mean of 75% as the efficiency of the turbines when developing their full power we have:

Turbine working with full capacity and using 565 cu. ft. of water per second: Head 125 ft., effective horse-power 6000, efficiency 75%.

Turbine using 75% of the water passed at full capacity, or 424 cu. ft.: Effective head 70.3 ft., head destroyed by throttling-gates 54.7 ft., effective horse-power 2160, efficiency corresponding to 125-ft. head 36%, while register or wicket gates would utilize the whole head of 125 ft. and give an efficiency of about 78%.

Turbine using 50% of the water passed at full capacity, or 283 cu. ft.: Effective head 31.25 ft., head destroyed by throttling-gates 93.75 ft., effective horse-power zero, efficiency corresponding to 125-ft. head, zero, while register or wicket gates would utilize the whole head of 125 ft. and give an efficiency of about 72%.

At first sight it seems absurd that a turbine using one half of the water that is required to give its full power should develop no power at all, but a few simple considerations will show this to be the case.[1]

[1] Meissner. Hydraulische Motoren, vol. 2, pp. 215 and 221.

To begin with, it should be stated that the size of the passage area of the guide- and runner-buckets is not changed by the position of the throttling-gates, and with half of the amount of water that is used at full load flowing through the buckets, the speed of the water in the buckets will also be only one half and the head corresponding to that velocity will be

$$H=\frac{(0.5c)^2}{2g},$$

or one quarter of the total head; the remaining three quarters of the head must be destroyed by the throttling-gates.

It will, therefore, at once be seen that the throttling-gate regulates the speed of a turbine by artificially changing the effective head, and these changes can only be in a downward direction, or by decreasing the head.

As a matter of fact, the same efficiency as obtained by throttling-gates can be had by regulating the turbines by means of the head-gates or tailrace gates, or by variable obstructions in the penstocks or draft-tubes, which latter, in fact, throttling-gates are.

It will also be seen that the turbine working under one quarter of the total head, and with only one half of the full amount of water, will give one eighth of the full power, or, in this case, 750 H.P., provided the turbine is allowed to run at the speed corresponding to one quarter of the total head, that is, at one half of its normal speed. However, when the speed of the turbine has to be kept normal, as required in almost every plant, especially when directly connected to an alternating dynamo, as in this instance, the water when flowing through the turbine at one half its proper speed will have no opportunity to do work in the turbine, but will drop freely through the runner-buckets.

This may be better understood if it is remembered that a turbine, no matter whether of the reaction or free deviation type, gives the best efficiency—and is therefore made to run—at about one half of the maximum velocity, due to the head, or, in other words, a turbine allowed to revolve freely will run with twice the velocity at which it gives the best efficiency; but the turbine can develop no power when running at the maximum

speed due to the head, as the runner-buckets move with the same velocity as the water and the water therefore can exert no pressure upon the runner-buckets.

These considerations will show that a turbine having a constant passage area of the buckets and running with a constant speed can develop no power when only one half of the full amount of water flows through the turbine, as the velocity of the water will also be only one half of the speed at full capacity or the speed of the turbine, remaining constant, will be twice as fast, relative to the water.

The turbines here considered will therefore, when running at normal speed but without any load except their own friction and the friction of the dynamos, directly coupled to the turbines, each require over 283 cu. ft. of water per second, or more than the amount required to develop 3000 H.P.

The wing- or throttling-gates are controlled by a hydraulic governor. The guide- and runner-buckets of the turbines are divided by a third crown into two unequal parts, and ring or cylinder gates, not shown in the figures, operated by an electric motor controlled from the switchboard, are provided outside of the guide-wheels. With these ring gates the smaller part of the guide-wheels can be closed and the turbine will then develop 4500 H.P., with practically full-gate efficiency, the arrarrangement being about the same as having two turbine units, the smaller one being shut down when the larger one alone is sufficient to carry the load, but the effect of the regulating- or throttling-gates on the efficiency is the same, except that when the ring gates are partly closed the throttling-gates are applied to a 4500-H.P. turbine.

The turbine-units can be shut down by fully closing the ring gates, the latter thus serving also as stop-valves.

Gages.—In nearly all modern engine-rooms of any importance will be found gages indicating the pressure in the steam-chest, receiver, and condenser, and similar gages used in connection with turbines enclosed in cases would be of immense value both to superintendents in charge of water-power plants, as showing them at once if the turbines are working properly, and to the hydraulic-power engineers and the turbine-builders, as giving

them valuable information concerning the proper proportions and arrangement of the different parts forming an installation.

Of such gages a pressure-gage, indicating the pressure of the water near the entrance of the guide-buckets, and a vacuum-gage, indicating the amount of draft or suction near the discharge-openings of the runner-buckets, would be the most important.

There should also be a gage showing the pressure between the runner-disk and the head of the case or the dome, and turbines having a thrust-chamber or thrust-piston should have a gage showing the pressure in this chamber or behind the piston.

At the lower end of a long penstock should be a gage indicating water-hammer and the rise and fall of pressure due to the speed regulation of the turbine.

A tachometer, or gage, showing at any moment at what rate of speed the turbine is running, and a dial with pointer indicating the gate-opening or position of the speed-regulating gates, will also be found of advantage.

CHAPTER VII.

GOVERNORS AND SPEED REGULATION.[1]

THE greatest difficulty encountered by the hydraulic-power engineer is the speed regulation of turbines under variable loads, and it has only been during the last few years that engineers have been able to regulate the speed of turbines supplied by long penstocks as closely as is common in steam-engineering practice. The reason for this will easily be seen if the conditions of the steam-engine and turbine regulations are compared.

Steam of 150 lbs. gage pressure weighs about 0.37 lb. per cubic foot, and the supply-pipe is seldom over 100 ft. long, so that the weight of the moving column of steam will rarely exceed 100 lbs., while the velocity is about 100 ft. per second. On the other hand, water weighs 62.3 lbs. per cubic foot and the penstock may be 1000 or even 10,000 ft. long, so that the weight of the moving column of water may be millions of pounds, while the velocity is rarely over 10 ft. per second. Thus the energy represented by the moving water-column may be hundreds or even thousands of times the energy represented by the moving steam-column.

Every change of load or power developed requires a change in the engine cut-off or in the gate-opening, of the turbine, and this in turn requires a change in the velocity of flow in the supply-pipe or penstock, which means a change in the amount of energy represented by the moving column.

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[1] For the principles involved in the speed regulation of turbines see "Elements of Design Favorable to Speed Regulation in Plants Driven by Water Power," by Allan V. Garratt, printed in the appendix to this book; also "Speed Government in Water-power Plants," by Mark A. Replogle, Journal of the Franklin Institute, Feb. 1898, p. 81.

Steam is compressible and elastic, and if the load of the engine, and thus the velocity required in the supply-pipe, is suddenly decreased, the excess of energy in the moving steam-column is absorbed by the compression of the steam contained in that column, while if the load and the required velocity in the pipe is suddenly increased the lack of energy is supplied by the expansion of the steam contained in the moving column. Both the absorption and the supply of energy are required only for a very short time, as owing to the small inertia of the steam-column the change in velocity is quickly attained. Water is incompressible and inelastic, and if the load of the turbine, and thus the velocity required in the penstock, is suddenly decreased, the excess of energy in the moving column must find some outlet, otherwise either the penstock or the turbine will be wrecked, while if the load and the required velocity in the penstock is suddenly increased the lack of energy must be supplied from some outside source. Both the escape and the supply of energy are required for a much longer time than in the case of steam, as owing to the great inertia of the water-column, the change in velocity can only be slowly attained.

Any decrease in the gate-opening and consequent decrease in the velocity of the water in the penstock will thus produce a temporary increase in the penstock pressure, and with the gates closing quickly, this increase in pressure may rise to the force and suddenness of a blow, usually called water-hammer. On the other hand, any increase in the gate-opening and in the velocity of the water in the penstock will produce a temporary decrease in the penstock pressure. Such changes in gate-opening will frequently cause long-drawn-out pulsations in the penstock pressure or surging of the water, and this action is often favored by badly arranged penstocks, relief-valves, stand-pipes, air-chambers, and connections, and aided by wave-motion and eddies in the headrace near the penstock entrance. It is evident that the increase and decrease in the penstock pressure, water-hammer, and the surging of the water, being in effect the same as varying heads acting on the turbine, must have a detrimental influence on the speed regulation of the turbine.

To obtain a good speed regulation it is often necessary, espe-

cially in connection with turbines supplied by long penstocks, to use some auxiliary device or devices for the escape and supply of energy or for the escape at least, which may be briefly considered in the following:

The pressure-relief valve serves for the escape of energy, when the pressure rises beyond a certain limit, in exactly the same manner as the safety-valve of a boiler. Relief-valves are well known, but nearly all of them are of poor design, being held closed by a single short helical spring of a few turns, and therefore can open only to a very small extent. A good spring relief-valve should have 3 to 6 helical springs, according to the size of the valve, and these springs should be long and have many turns, so as to permit the valve to open sufficiently, without requiring too great a rise in pressure in the penstock. Provision should be made for ascertaining at any time whether the valve is in working condition and set for the proper pressure. This can usually be done by a lever-and-chain attachment for pulling the valve open or by slackening the springs, while the valve is under water-pressure, until the valve blows off, and to avoid the drenching of the man testing the valve, the latter should be enclosed like the pop safety-valves for boilers.

The Lombard pressure-relief valve, shown diagrammatically in Fig. 59, is a great improvement over the ordinary relief-valve. Its action is as follows: *A* is the end of a penstock or a nozzle of a penstock in which the pressure is to be relieved when a certain limit has been reached. The disk of the relief-valve *c* is held to its seat against the water-pressure in the penstock by the water-pressure behind the piston *e*, the pressure-water behind the piston being supplied from the penstock through pipe *f*. As the pressures per unit area against the valve-disk and behind the piston are equal, the piston is made larger in diameter than the disk, so that the total pressure behind the piston will not only overcome the total pressure against the valve-disk, but also hold the latter firmly to its seat. The space behind the piston *e* is also connected through pipe *i* to the waste-valve *D*. This is a balanced valve held closed by means of the spring *p*, while the water-pressure in the penstock, communicated through pipe *r* and acting behind the piston *n*, tends to open the waste-valve. The force of the

spring p can be regulated so that the water-pressure will overcome the force of the spring and open the valve at any desired pressure in the penstock.

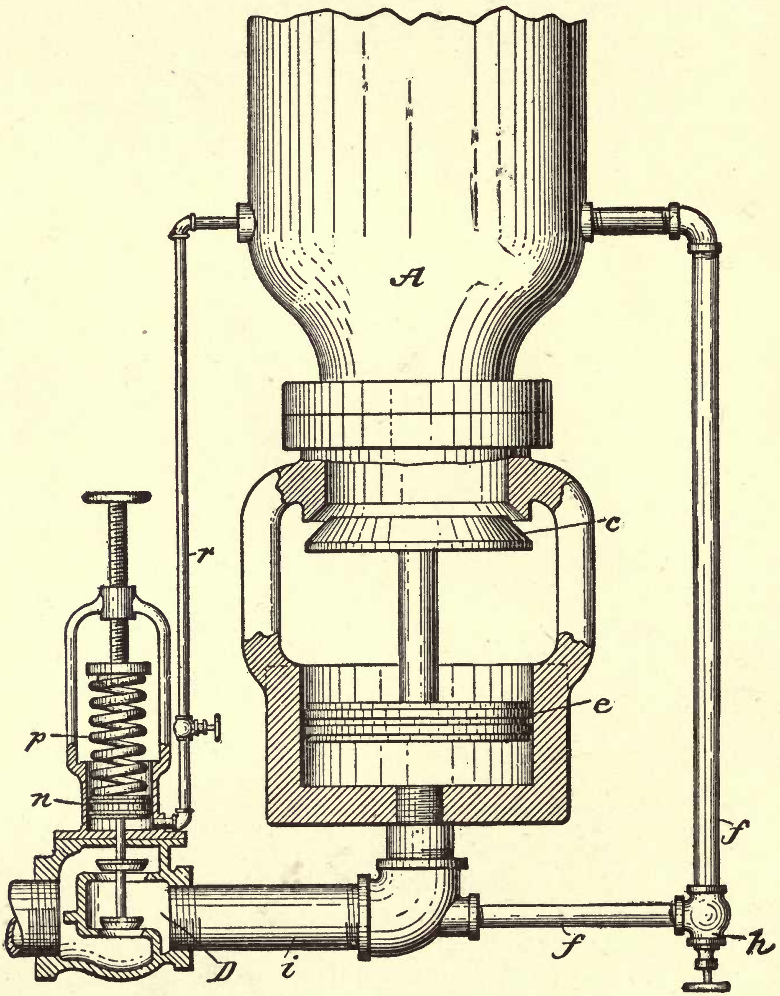

FIG. 59.—Lombard Pressure-relief Valve. Made by The Lombard Governor Co., Boston, Mass.

With the relief-valve closed and the water-pressure in the penstock rising above the normal to the pressure for which the

spring *p* is set, the piston *n* will open the waste-valve, which will relieve the pressure behind the relief-valve piston *e* and allow the pressure-water to escape. While water will begin to flow through pipe *f* as soon as the waste-valve is opened, yet the area of pipe *f* is so much smaller than the area of pipe *i* and the waste-valve opening, that the pressure behind the piston *e* will at once fall below the pressure which exists in the penstock, and therefore the pressure in the penstock forces the relief-valve disk *c* and piston *e* back, or, in other words, opens the relief-valve.

The greater the rise in pressure is in the penstock the greater will be the extent to which the waste-valve opens, and consequently the greater will be the reduction in pressure behind the piston *e*, and therefore the greater the extent to which the relief-valve opens.

As soon as the pressure in the penstock has fallen to the pressure for which the spring *p* is set, the latter closes the waste-valve, the pipe *f* restores the full penstock pressure behind the piston *e*, and the latter closes the relief-valve. To prevent any surging in the penstock, due to the closing of the relief-valve and the consequent retardation of the water, the relief-valve is made to close slowly, the rate of closing being adjustable by means of the valve *h*.

Relief-valves must be prevented from freezing or from becoming incrusted with ice, as otherwise they may be rendered entirely useless.

The by-pass, which may be employed where economy in water consumption is not demanded, consists of a valve or gate of sufficient area to pass the entire volume of water required by the turbine at full-gate opening and moved in conjunction with the speed-regulating gates. With the turbine-gate fully open, the by-pass is closed, but when the regulating-gates commence to close, the by-pass opens and its passage area is increased in the same proportion as the gate-opening decreases, so that the combined area of the gate-opening and the by-pass is always sufficient to pass the entire volume of water required by the turbine at full-gate opening; thus the velocity of the water in the penstock and the amount of water discharged remains always the same, the discharge of the by-pass being run to waste. This arrangement not only permits the closest speed regulation with violently fluc-

tuating loads, but also relieves the penstock from shock or water-hammer, and is therefore often used in connection with impulse turbines working under high heads and supplied by very long penstocks.

European engineers have abandoned the ordinary by-pass on account of the great waste of water which its use implies, but frequently use the temporary by-pass, which is essentially the same device as the ordinary by-pass, but the speed-regulating gates or their rigging and the by-pass are connected by means of a dash-pot or cataract. The temporary by-pass will thus open while the speed-regulating gate closes, in the same manner as described in connection with the ordinary by-pass, but as soon as the closing movement of the regulating-gate ceases, the by-pass at once starts automatically to slowly close again, being actuated by a spring, counterweight, or hydraulic pressure. The speed with which the by-pass closes is easily regulated by changing the size of the aperture connecting the two ends of the dash-pot. The temporary by-pass does not open at all when the regulating-gate closes very slowly. It will be seen that the temporary by-pass is similar in its effect to the relief-valve, except that the by-pass opens before a rise in the penstock pressure, due to the closing of the regulating-gates, takes place.

The temporary by-pass gives full protection to the penstock and permits the closest speed regulation with a decreasing load, but cannot, of course, assist the governor to prevent slowing down of the speed when the load on the turbine is increasing.

A by-pass is usually located outside of the turbine-case, so that its discharge will not interfere with the proper working of the turbine or draft-tube.

The stand-pipe is frequently employed to aid the governor and thus to improve the speed regulation of turbines, and is simply an open reservoir which, to a limited extent, will absorb or store energy, when the gate-opening is decreased in consequence of a reduction in the load of the turbine and will supply energy when the gate-opening is increased in consequence of an increase in the load of the turbine. The stand-pipe is the best possible relief-valve and should have its top edge a few feet above the high-water level in the headrace and its diameter or capacity should be in

accordance with the volume of water discharged by the turbine at full-gate opening and the length of the penstock.

When the gate-opening of the turbine is suddenly reduced, the excess of water flows into the stand-pipe, causing the water therein to rise, and perhaps to escape over the top edge of it, until the water-column in the penstock has slowed down, while when the gate-opening is suddenly enlarged the additional water required is supplied from the stand-pipe, causing the water therein to fall until the speed of the water-column has increased to meet the demand. In connection with high heads stand-pipes are rarely used, as they are not only very expensive but also less effective on account of the inertia of the water-column in the stand-pipe.

Stand-pipes must be carefully protected from freezing, as otherwise they may be rendered entirely useless. A waste-pipe should be provided to carry off the water escaping over the top edge of the stand-pipe.

Air-chambers are often used on penstocks, but while they may be useful to protect the penstock against the effects of water-hammer, they are of little or no value as an aid to the regulation of the turbines. To cushion the shocks in a penstock an air-chamber should be of ample capacity, and as air is readily absorbed by the water, an air-pump should be provided to replace the air thus carried off. Gage-glasses and try-cocks should be placed on each air-chamber, so that it may at once be seen whether it is effective.

The blows struck by water in a penstock or water-hammer are first and most violently felt at the lower end of the penstock and in the direction in which the water-column moves, and from there back up, so to say, with diminishing strength towards the upper end of the penstock. Therefore all such safety devices as the relief-valve, by-pass, stand-pipe, and air-chamber should be at the extreme lower end of the penstock and their discharge or connection should be in the direction in which the water-column moves. A stand-pipe should be connected with the penstock by a short, straight pipe of large diameter and an air-chamber by a short neck, also of large diameter.

The fly-wheel is frequently employed in Europe, and to some extent in America, to aid the governor and thus to improve

the speed regulation, especially in connection with turbines working under high heads, but a fly-wheel cannot, of course, protect a penstock against water-hammer. A turbine-runner has very little fly-wheel capacity, and the use of a fly-wheel will therefore eliminate the small variations in speed, due to slight but sudden fluctuations in load, to water-hammer, to the surging of the water in the penstock and draft-tube, and to other causes, which momentary variations even the best governor cannot prevent. The amount of energy which a fly-wheel can absorb or give out is only small, but it will at least retard the changes in speed of the turbine with changes of load. Where the turbines are used to drive dynamos sufficient fly-wheel capacity may be given to the armature or revolving field to make a separate fly-wheel unnecessary. This plan was adopted by the Niagara Falls Power Co. for its 5000-H.P. alternating machines.

Turbine-gates, owing to their large size and weight and the great resistance offered by the water and by friction, require a large amount of power for their movement, and for this reason cannot be actuated directly by a sensitive centrifugal governor. An auxiliary machine, called a relay or servomotor, and controlled by the centrifugal governor, is therefore employed to actuate the gates, and the governor is called, according to the form of power used in the relay, a mechanical or hydraulic governor.

Mechanical and hydraulic governors of good design both serve their purpose equally well, but a hydraulic governor is usually less complicated in construction than a mechanical one. In Europe, governors with electrical and pneumatic relays have also been tried, but have not proven a success.

A governor occasionally used for small turbines is the brake-governor. With this governor the speed-regulating gate remains fully open and the turbine thus always develops its maximum power, the power in excess over the demand being destroyed by a friction resistance.

A modern turbine governor consists of three principal parts, viz.: the centrifugal governor, which may be of any approved

design; the relay, using either mechanical or hydraulic power and controlled by the centrifugal governor, and the return.[1] When a change of load and consequent change in the speed of a turbine takes place, the governor will set the regulating-gate in motion, but owing to the inertia of the water a certain amount of time is required for the turbine to return to the proper speed, and the regulating-gate would have traveled in the meantime beyond the required position and would have to start at once in the opposite direction; that is, the turbine would be overgoverned or racing if the return were not used to arrest the motion of the relay and with it the motion of the regulating-gate before the latter has traveled too far.

Of great importance in connection with turbine governors is the time of closing—that is, the time required by a governor to move the regulating-gate from the fully open position to the fully closed position—as it will easily be seen that the quicker the governor moves the gate the closer will be the speed regulation. A mechanical governor will usually require from 15 to 25 seconds for an entire closing movement of the gate, although mechanical governors have been built which required only 3 seconds. With hydraulic governors the time of closing may be reduced to one second, and hydraulic governors will therefore give the closest possible speed regulation.

To show the performance to be expected from a modern turbine governor, it may be stated that with favorable conditions the speed may be kept within 5 or 6% of the normal when the full load is suddenly thrown off and it will require between five to fifteen seconds for the turbine to return to normal speed. With ordinary electric-railway loads, speed variations of about 3% as a maximum may be expected.[2]

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[1] The reference letters given in Figs. 28, 32, 33, 48, 52, and 66 to 69 have the following meanings: *C* is the centrifugal governor, *M* is the relay, *S* is the valve actuated by the centrifugal governor and controlling the relay, *Z* is the return, and *R* is the connection between the relay and the regulating-gate.

[2] The figures here given are taken from "Elements of Design Favorable to Speed Regulation," etc., by Allan V. Garratt, printed in the appendix to this book.

The mechanical governor, at one time the only turbine governor obtainable, has been replaced to a great extent by the hydraulic governor. The power for a mechanical governor is usually taken from the turbine which it regulates and is transmitted to the gate by means of a ratchet-wheel and pawls, friction gearing or clutches.

The mechanical governor best known in America is the Replogle governor. The centrifugal governor and the return of a Replogle governor, employing electric current for throwing the relay into or out of action, is shown diagrammatically in Fig. 60 and may be described as follows:[1]

In Fig. 60, *Z* is a speed governor; *A* is a vertically sliding bracket which is supported by cam *B*; *S* is a toothed segment operated by the turbine-gate shaft *G*, which is also toothed; *C* is an electric contact-point supported by a lug in the top of *A*; *O* is a smilar contact supported by a bottom lug; *L* is the lever tilted by the governor-balls when a variation of speed occurs (it serves also to complete the battery circuit by touching contacts *C* or *O* when the speed varies); *D* is a binding-post through which the battery current enters the lever *L*.

Bracket *A*, contacts *C* and *O*, lever *L*, cam *B*, segment *S*, which operates the cam, and gate-shaft *G* with its pinion are known as the relay controlling device, and assuming the turbine to be running at normal speed their operation may be explained as follows: If additional load is thrown onto the turbine, the governor-balls will allow lever *L* to touch contact *O* (which must be connected by a wire conductor to magnet *O*. It is assumed that the switch *E* is closed). Magnet *O* being thus energized, trips the auxiliary machine or relay, not shown, which starts to open the turbine-gates by turning *G* in the direction of the arrow. *G* in turning carries the segmental rack *S* with it, which lowers cam *B*, allowing bracket *A*, consequently contact *O*, to drop away from lever *L*, breaking the electrical circuit, which cuts the auxiliary power out of action and stops the gate motion before the added power has given any increased velocity to the turbine. If the

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[1] The description of this governor was kindly furnished by Mr. Mark A. Replogle.

added water is enough to carry the new load no further action of the governor will occur. If the speed continues to drop this operation will be repeated until the speed will not drop farther.

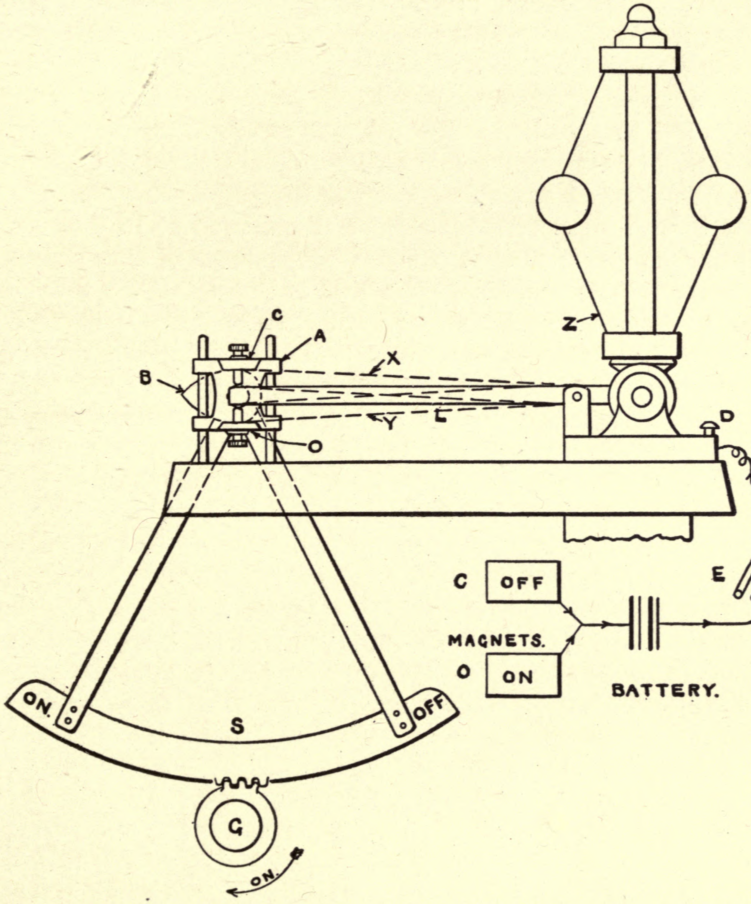

FIG. 60.—Centrifugal Governor and Return of Replogle Governor. Built by The Replogle Governor Works, Akron, Ohio.

(It must be remembered that the lever *L* varies its position with the slightest variations in speed.) Since the contact end of *L* is now maintained in a lower position it follows that the speed is slightly lower.

There are two reasons for this complex action of the relay controlling device: (1) The effect of the added water is not available until some little time after the turbine-gates are opened. Therefore it is necessary to stop the gate motion before the speed has increased, as the added water may cause too great an increase in speed, which will cause a "see-sawing" or racing effect. (2) The momentum or fly-wheel capacity of the plant must always carry the added load until the new power is furnished by the water. In order to draw this power from the revolving parts their speed must be dropped enough to feed out the proper amount. The cam *B* is designed or made steeper or flatter so as to feed out exactly the proper amount of power in opening gate or to absorb it in the reverse operation. If a load is thrown off the turbine the operations are all reversed. This allows lever *L* to have position *X* when the turbine has no load and the position *Y* when the turbine is carrying full load. This variation is equal to from 1 to 6% in speed, depending on the amount of momentum or fly-wheel effect in the plant. If the fly-wheel effect is ample a flat cam of 1% variation can be used. If 1% drop in speed will not feed out storage power enough to carry the load until gravity acts then a larger drop must be resorted to or the result will be hunting or racing.

In a later mechanical governor, Replogle has added another refinement as follows: After a change in load has been balanced by the above principles, a set of springs aided by a cataract cylinder gradually returns the relay controlling apparatus to its original central position, turning on or off water slowly until the speed is exactly normal although the turbine-gate sets at a different position. This is called a "Relay Returning Governor."

A still later governor has a further refinement which embraces all of the actions referred to above and an additional action that gradually increases the speed as the load goes on.[1] With this governor it is possible to have the speed automatically increase from normal speed to as much as 10%, from no load to full load, and yet all of the previous operations have taken place at each change in load. The idea of this over-returning-governor is to

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[1] See Engineering News, Nov. 13, 1902, p. 409.

make up for line losses in direct-current transmission or to offset belt lag when belt or rope drives are used. This same governor can operate as either the first or second styles mentioned by simple adjustments, and is called "Replogle's Differential Relay Governor."

One great drawback in turbine-governing is that sufficient fly-wheel effect is not always furnished in the plant construction. Often there is practically none, even when other conditions are bad, requiring a governor to be a complex machine.

The hydraulic governor, although having been in use only for a few years, has found an extensive application.[1] The power of a hydraulic governor is furnished by the pressure of a fluid acting behind a piston in the relay-cylinder. The pressure fluid for hydraulic governors used in connection with turbines working under heads of about 200 ft. or more is, as a rule, water taken directly from the penstock. To avoid carrying over sand or gritty matter from the penstock to the relay, which would cause rapid wear of the working parts of the relay, the water connection for the latter should be made by means of a nozzle of large diameter and having a length of about twice the diameter. This nozzle is set vertically on top of the penstock, in a location where water-hammer is the least likely to affect the relay. The nozzle is closed at the end by a head or blank flange and the supply-pipe for the relay connected to the centre of this head or blank flange. Thus with a supply-pipe 2 ins. in diameter and a maximum speed of water in this pipe of 4 ft. per second, the upward movement of the water in a nozzle 24 ins. in diameter would never be greater than about $\frac{1}{3}$ in. per second, or far too small to carry any sand or gritty matter along with it.

In most cases, however, it will be found advisable to pass the pressure-water used in a hydraulic relay through a filter or fine screen or sieve to keep out any light or floating matter. Two filters or screens should be provided, so that one may be cleaned while the other is in operation.

A simple water-filter is shown in Figs. 61 and 62. The water

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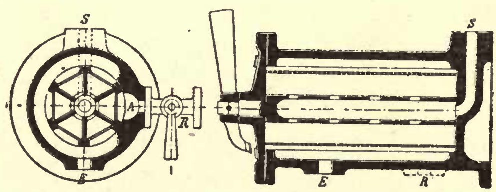
[1] The hydraulic governor was invented in 1885 by Piccard, Pictet & Co., Geneva, Switzerland.

enters the filter through the opening *E* and leaves through the opening *S*. The screen drum has six compartments, of which the one opposite the chamber *A* is not in use and may be cleaned by opening the valve *R*, which causes the filtered water from the interior of the drum to flow through the screen in the reverse direction, thus washing off any matter clogging the screen. By rotating the drum one screen after the other may be washed.

The interior working parts of a relay using water as a pressure-fluid are frequently lined with brass to prevent corrosion, which latter might cause the relay to stick.

Where the head under which the turbine is working is less than about 200 ft., or where it is desirable for other reasons, the



Figs. 61 and 62.—Water-filter for Hydraulic Governors. Built by Escher, Wyss & Co., Zurich, Switzerland.

pressure-fluid employed for the hydraulic governor is usually oil, though if the pressure-fluid is liable to be exposed to freezing temperatures, a mixture of glycerine and alcohol is often employed. The oil used should be of comparatively high viscosity, absolutely non-corrosive, have a low freezing-point, and should not break down into vaseline after long service.

To produce the necessary pressure an oil force-pump is required, and to equalize the supply and demand of oil an accumulator must be provided, which should be of the receiver type, that is, a closed tank, or receiver, partly filled with oil, which is subjected to air-pressure. A weighted accumulator is not to be recommended on account of the inertia of the weights. Where the head is sufficiently high the accumulator may be dispensed with by subjecting the oil to the pressure of the water in the penstock.

The use of oil as pressure-fluid has the advantage that the interior working parts of the relay require no separate lubrication and the wear of these parts is reduced to a minimum. With the

proper kind of oil the same pump may be used to supply the relay and to furnish a forced oil lubrication for the turbine bearings; thus the oil used for the collar-bearing of the turbine shown in Figs. 22 and 23 is supplied by the relay pressure-pump.

The hydraulic governor best known in America is the Lombard governor.[1] An attachment may be used in connection with the Lombard governor, by which the speed of a turbine can be varied while the latter is running; thus a turbine driving an alternating dynamo may be brought up to speed and the alternator synchronized to throw it into parallel. For this purpose the stem of the valve controlling the relay is made in two parts, which are connected by a sleeve-coupling with right- and left-hand threads. By turning the coupling the valve-stem is lengthened or shortened, and thus the position of the valve in reference to the centrifugal governor is changed. The coupling may either be turned by hand or by a small electric motor controlled from the switchboard.[2]

The Voith hydraulic governor is also provided with a device by which the speed of the turbine can be varied while the turbine is running.[3] For this purpose the step upon which the spindle of the centrifugal governor revolves may be raised or lowered and thus the position of the centrifugal governor changed in reference to the valve controlling the relay. The raising or lowering is done by a screw spindle, which may either be turned by hand or by a small electric motor controlled from the switchboard.

In Fig. 63 is shown a simple and efficient hydraulic governor applied to an impulse turbine. Its action is as follows: In the valve-chamber g is the admission-valve r, which is in the shape of a differential plunger and is kept floating by the difference in water-pressure existing in the spaces a and b. The pressure-water after passing through the filter enters the valve-chamber at a and flows through the bore in the admission-valve into the space b. The pressure in space b is regulated by the size of the vent-hole opening, which is controlled by the regulating-valve v, which in

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[1] Built by The Lombard Governor Co., Boston, Mass. For an illustrated description see Frizell, Water Power, p. 575.

[2] See Engineering News, Jan. 15, 1903, p. 62.

[3] Built by J. M. Voith, Heidenheim, Germany. For an illustrated description see Zeitsch. d. V. deutsch. Ing., June 20, 1903, p. 894.

turn is moved by the centrifugal governor. At normal speed the admission-valve *r* is in the middle position as shown, the port *d* of the relay cylinder being closed. Water flows continuously from *a* through the bore in the admission-valve *r* to the space *b*

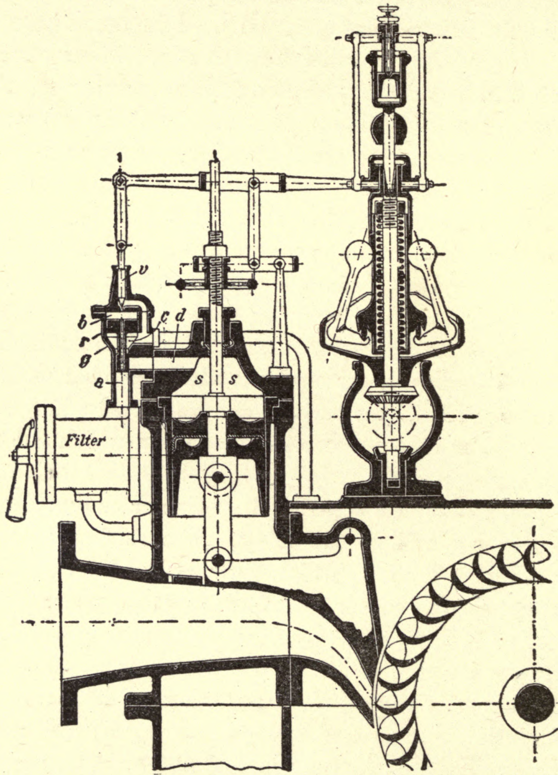


FIG. 63.—Hydraulic Governor for Impulse Turbines. Built by Escher, Wyss & Co., Zurich, Switzerland.

and from there through the vent-valve to the space *c* and then escapes through the waste-pipe shown.

When the speed of the turbine decreases, the centrifugal governor raises the regulating-valve *v*, the pressure in the space *b* is reduced, and the admission-valve raised, permitting the pressure-water to flow from *a* through the port *d* into the relay-cylinder, forcing down the relay-piston and thus opening the tongue of

the turbine-nozzle, that is, increasing the gate-opening. However, as the fulcrum of the lever which moves the regulating-valve *v* is connected to the piston-rod of the relay-piston, the downward movement of this piston returns or lowers the regulating-valve *v*, which decreases the vent-hole opening and thus causes an increase in the pressure in the space *b*, which forces the admission-valve back to its middle position and thus stops the motion of the relay-piston and the tongue. With an increase in the speed of the turbine, the opposite action takes place, the regulating-valve *v* being lowered and the pressure in the space *b* is increased. This forces the admission-valve *r* down and permits the pressure-water in the relay-cylinder to escape through port *d*, space *c*, and the waste-pipe shown; and the water-pressure existing in the space above the tongue and below the relay-piston forces the latter upwards.

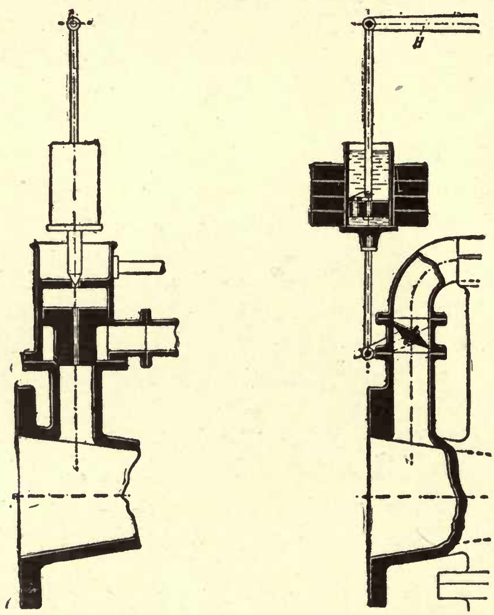
From the foregoing description it will be noted that for each position of the relay-piston and the tongue or gate, there is a certain corresponding position of the fly-balls of the centrifugal governor and consequently a certain corresponding speed of the turbine.

This governor is very sensitive and quick-acting on account of the small mass and inertia of the regulating-valve *v*, the comparatively large areas of *a*, *b*, and *c*, the floating state of the admission-valve *r*, and the continuous flow of water through the valve-chamber *g*.

In Figs. 64 and 65 are shown two forms of a temporary by-pass often used in connection with and controlled by a governor.

The action of the by-pass shown in Fig. 64 is very similar to that of the admission-valve of the governor above described. The valve is in the shape of a differential plunger and has a hole bored through its centre, permitting the water to flow to the space above the plunger. The pressure in the space above the plunger, which holds the by-pass closed, is controlled by a conical valve, which receives its motion from the piston-rod of the relay-piston. With a rise in the speed of the turbine and the consequent reduction in the gate-opening, the conical valve is raised and thus the pressure above the plunger relieved and the plunger forced upward by the pressure below it; that is, the by-pass is

opened, the extent to which the by-pass is opened being in proportion to the extent to which the gate-opening is reduced. The dash-pot shown above the conical valve, by its own weight gradually lowers the conical valve again, thus permitting the pressure above the plunger to rise and to close the by-pass. The by-pass is of such a size that when necessary the entire volume of water



FIGS. 64 and 65.—Two Forms of Temporary By-pass. Built by Escher Wyss & Co., Zurich, Switzerland.

required by the turbine at full gate-opening may be discharged through it.

The wing-valve of the by-pass shown in Fig. 65 is opened by the lever *H*, which receives its motion from the piston-rod of the relay-piston and the weighted dash-pot gradually closes the by-pass again.

The hydraulic governor described above is used for regulating the speed of the double impulse turbine shown in Figs. 54 to 56. The by-pass is located below the nozzles and the dash-pot on

the right side of the turbine, as shown in Fig. 54, the piston of the dash-pot forming the counterweight for closing the by-pass.

A hydraulic governor having a combined relay and by-pass and using oil under water-pressure from the penstock as a pressure-fluid is employed in the regulation of the impulse turbine shown in Figs. 51 to 53. The centrifugal governor, relay, and return act in the same manner as above described, but the piston of the relay serves at the same time as the cylinder for a second piston *L*, which moves inside of the first one and actuates the by-pass. The by-pass *P* itself is similar in shape to the turbine-nozzle and is opened and closed by a slide fastened to a pair of bell-cranks, which are connected to the by-pass piston and whose fulcrum is in line with the axis of the hinge of the tongue. The slide and its seat at the end of the by-pass nozzle have a cylindrical surface. With the turbine running at normal speed, a helical spring surrounding the piston-rod of the relay-piston holds the by-pass slide closed and resting against a stop. A small aperture admits oil to the space between the relay and the by-pass pistons, being sucked into this space, when the space is enlarged by the relay-piston moving upwards in reference to the by-pass piston.

With an increase in the speed of the turbine and consequent downward movement of the relay-piston and the tongue—that is, a reduction in gate-opening—the oil contained in the space between the relay and by-pass pistons forces the by-pass piston also downward, thus opening the by-pass slide, but the pressure of the spring gradually moves the by-pass piston back to its upper position, thus closing the by-pass and forcing the oil contained in the space between the relay and by-pass pistons out through the small aperture, the speed with which the by-pass closes depending on the size of this aperture.

This governor keeps the speed of an impulse turbine as shown in Figs. 51 to 53, working under a head of 1312 ft. and supplied by a penstock 6560 ft. long, within 3% of the normal when the full load of several hundred horse-power is suddenly thrown off and the maximum variation in pressure in the penstock is only 10%.

Figs. 66 to 68 show a hydraulic governor with differential relay-piston. This governor is used in the Elektrizitaetswerk

Beznau, at Beznau, Switzerland, a cross-section of the power-house of this plant being shown in Fig. 19.

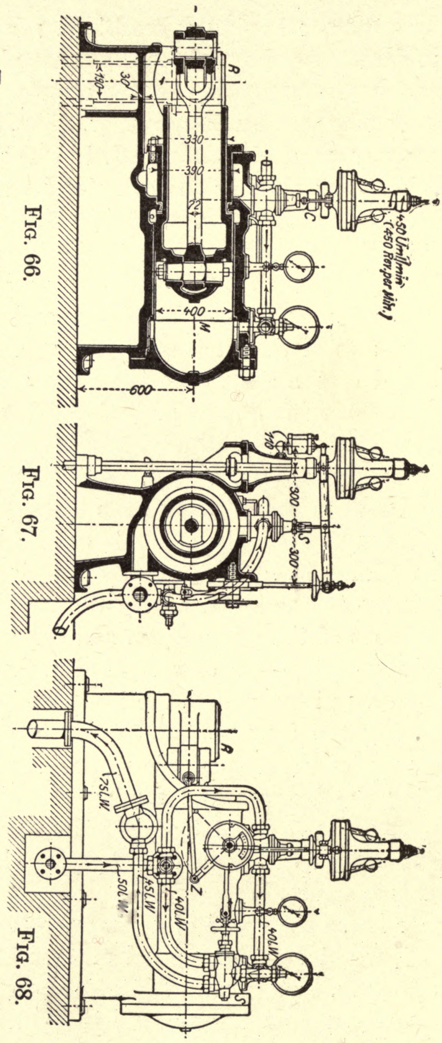


Fig. 66.

Fig. 67.

Fig. 68.

Figs. 66 to 68.—Hydraulic Governor. (Built by Th. Bell & Co., Kriens, Switzerland.)

Fig. 69 shows a very simple hydraulic governor in which two plungers are employed for the relay, instead of a piston.

The hydraulic governor shown in Figs. 32 and 33, in connection with the turbine for the regulation of which it is used, keeps the speed of this turbine within $1\frac{1}{2}\%$ of the normal, with variations in load of 10%, and within 5% of normal speed when the full load is suddenly thrown off. However, a small fly-wheel is used to aid the governor.

Tests made with the hydraulic governors used in power-house No. 2 of the Niagara Falls Power Company gave a maximum

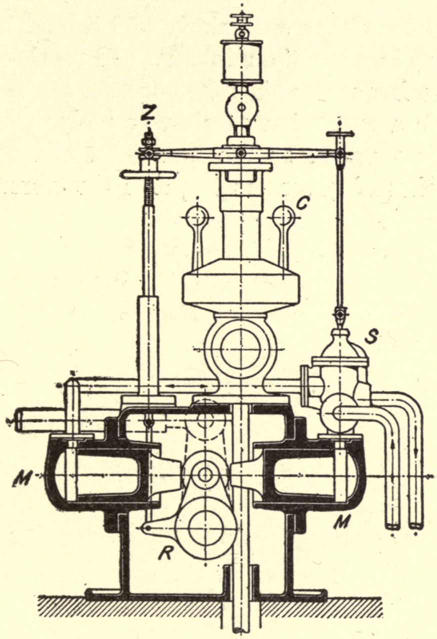


FIG. 69.—Hydraulic Governor. (Built by Escher, Wyss & Co., Zurich, Switzerland.)

variation from the normal speed of the turbines of 3.8% when the full load of 5000 H.P. was thrown off as rapidly as possible.

In many large steam-power plants one steam-engine takes care of the regulation of the whole plant, and in an analogous manner one turbine unit and governor may be used to take care of the regulation of a large water-power plant.

Builders of high-class steam-engines always make the governor

a part of the engine, but turbines and turbine-governors are at present entirely separate machines, and the governor is usually set anywhere about the plant; yet it would be well for the turbine and governor builders to combine, or at least to cooperate, and to mount the governor on the main frame or the turbine-case, thus making the whole a complete and self-contained machine, as is now the general practice with European turbine-builders.

In contracting for a governor, it should be clearly stated whether the guaranteed limit in speed variation of the turbine to be regulated is the variation either way, above and below the normal, or the total variation between the minimum and maximum speed.

The guarantee frequently given by governor-builders, that their governor will give a regulation as good or better than any other governor, is of no value, as the purchaser can evidently not test all other makes of governors to prove the correctness of the guarantee.

PART II.

WATER-POWER PLANTS

CHAPTER VIII.

WATER-CONDUCTORS.

Headrace and Tailrace.—Narrow and deep headraces and tailraces are always to be preferred to wide and shallow ones, as the loss in head is less in a deep than in a shallow race, but the greatest advantage from a deep race is derived in localities where thick surface ice is formed during the winter months, as such ice not only greatly reduces the passage area of the race, but also offers considerable frictional resistance to the flow of the water.

The location and direction of the entrance to a headrace should be carefully chosen to avoid, as far as possible, the tendency of the water to carry sand into the headrace. Across the entrance of a headrace should be placed a heavy boom, to prevent wood, ice, and floating rubbish from entering the race. A crib with water-passages near the bottom is an improvement over the boom, and should be arranged for the use of stop-logs, to shut out the water from the headrace when desired.[1] A boom or crib should have such a direction that floating matter will not lodge against it but will glance off and be carried away by the stream.

All bends should be avoided in races, but where necessary these bends should be of long radius.

[1] For an illustrated description of such a crib see Engineering News, May 7, 1903, p. 400.

In American practice the speed of the water in a head- or tailrace is usually 2 to 3 ft. per second, while in Europe a speed of 1.5 ft. per second is mostly employed. In cold climates the speed of the water in the races should be sufficiently low, that is not over 3 ft. per second, so as to allow the water to freeze over and thus prevent the formation of anchor-ice and frazil in the races.

Water carrying much sand is frequently run through a sand-settler, if the volume of the water is small. Such a sand-settler is a basin or an enlargement in the headrace, through which the water flows with so low a speed as to permit the sand to be precipitated. Grooves or other obstructions are placed on the bottom and at right angles to the direction of the flow of the water, to arrest sand rolling along the bottom. Sand-settlers employed in connection with open timber flumes are usually large, shallow wooden boxes. To free large volumes of water from sand, a ditch is often placed either in front or back of the water-racks, in which at least the sand rolling along the bottom will be caught. By means of an 18- or 24-in. pipe, closed by a gate and leading to some place outside of the headrace, the sand accumulated in the ditch may occasionally be washed or scoured out. The inlet of the pipe should be protected by a coarse screen, to prevent large objects from entering and lodging in the pipe. As a rule it will be found cheaper to let the sand go through the turbines, and to renew the guides and runners when worn out, than to attempt to free large volumes of water from sand.

Near the lower or power-house end of the headrace should be placed a large sluice-gate and wasteway, to discharge ice and floating rubbish from the headrace, and a small boom may be used to guide such floating matter to the gate.

The tailrace under the turbines should be arranged to give the water discharged every facility for escaping, to prevent it from backing up around the turbine or draft-tube. The walls of the tailrace should be of such a shape as to deflect the water in the proper direction, but where draft-tubes are used the best arrangement is to have the draft-tubes curving or inclined in the direction of the flow of the water in the tailrace. Where several draft-tubes discharge into the same tailrace it is advisable to place these draft-tubes on one side, as they will thus cause less

obstruction to the passage of the water than when placed in the centre of the tailrace.

In localities where the winter is severe the part of the tailrace located under the power-house should be protected against extreme cold, to prevent an excessive accumulation of ice. For this purpose the upper part of the tailrace opening, that is the end of the tailrace where it emerges from under the power-house, should be boarded up to within 1 or 2 ft. of the normal tailwater level, and to the bottom of this wooden partition should be nailed a strip of canvas or tarpaulin of such a width that its lower edge will float on the tailwater, even at its lowest stage. A board hinged to the bottom of the wooden partition and reaching to the level of the tailwater at its lowest stage may be used instead.

Water-racks.—Water-racks will give reasonable safety against choking or damaging the turbines if the space between the bars is less than the least clear dimension of the water-passages in the guide- or runner-buckets. Temporary choking may occur with the turbine-gates nearly closed, but this can usually be relieved at once by opening the gates. This rule may be employed with the European type and small sizes of the American type of turbine, but the larger sizes of the latter give too great a spacing.

Coarse racks are sometimes used in front of the fine racks, especially where the fine racks are not protected by a boom or crib. For size and spacing of the rack-bars, the following figures are considered as good practice:

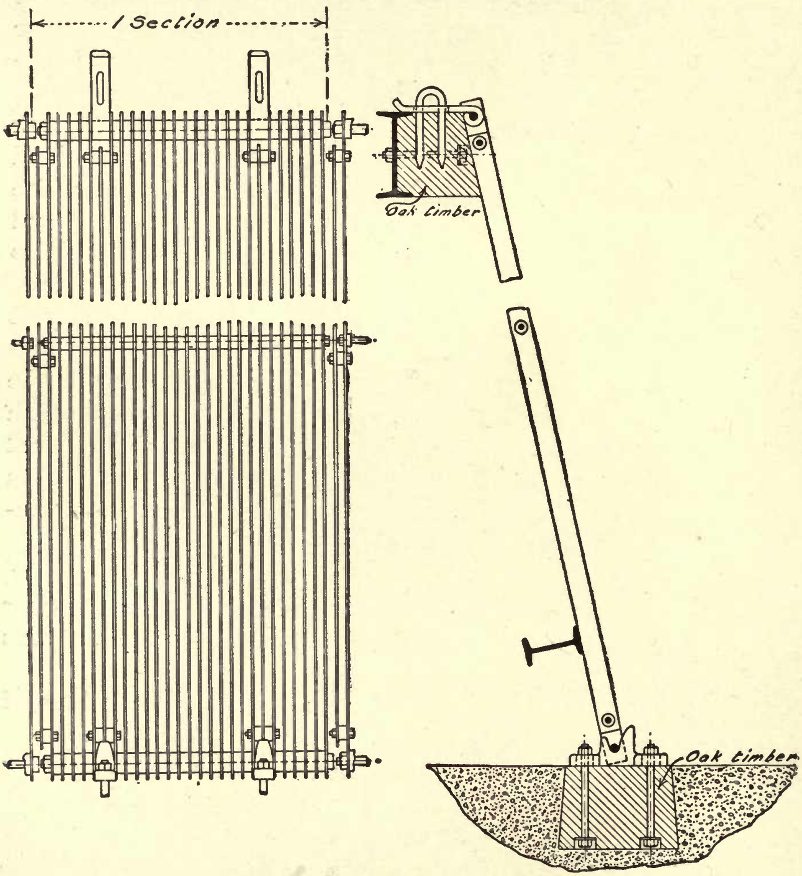
Fine racks: Clear space between bars $\frac{3}{4}$ to $1\frac{1}{2}$ ins.; bars of wrought iron or steel, $\frac{1}{4}$ to $\frac{3}{8}$ in. thick by 3 to 4 ins. wide.

Coarse racks: Clear space between bars 3 ins.; bars of wrought iron or steel, $\frac{1}{2}$ to $\frac{3}{4}$ in. or even 1 in. thick by 4 to 5 ins. wide.

Water-racks should have a total clear area for the passage of water much in excess of the total area of the penstock inlets, to permit the passage of the water without loss of head, even with the rack partly clogged. To give a large passage area and to facilitate cleaning, the racks are usually inclined. In calculating the strength of the supporting structure for a water-rack, it is customary to assume some arbitrary pressure against the rack, say between 30 to 50 lbs. per square foot.

Instead of a continuous rack, firmly fixed in position, a sectional

rack, as shown in Figs. 70 and 71, should be used. The sections are made 3 to 4 ft. wide, so that they may be easily handled, and are held at the lower end by a pair of cast-iron shoes, and at the



FIGS. 70 and 71.—Sectional Water-rack.

upper end by clasps or latches or similar means, so that any section can be taken out, repaired, or the bars straightened and put back into its place, without the employment of a diver. In cold climates such racks have to be securely held, as with the surface ice frozen to the bars any rise in the water-level would lift the racks from their shoes.

Head-gates.—The head-gate is used for shutting off the water from a penstock, open turbine-chamber, or forebay, and in America is nearly always a slide moving vertically. Until recently head-gates were made of wood, but now they are frequently made of steel plate and structural steel, especially if of large size. Sliding gates are moved either by means of racks and pinions or screw-spindles, operated by hand, or mechanical or electric power, or by means of hydraulic lifts. Gates that are likely to get out of line while being moved, and thus to bind or jam in their slides, should be provided with a squaring-shaft.[1] Such a squaring-shaft is particularly required where two hydraulic lifts are used for one gate, as otherwise the piston of one lift might move faster than the other one.

For large sliding head-gates a by-pass or balance-port is usually employed, by which the water-pressure in front and back of the gate can be balanced before moving the gate.

In Figs. 72 to 75 is shown a sliding steel head-gate provided with a balance-port. The gate is made in two sections, the upper section being much smaller than the lower one. The horizontal joint-faces, where the sections meet, are planed. The gate-stem is fastened to the upper section only, and the two sections are connected by two rods, which, however, leave the upper section free to move 12 ins. without moving the lower section, so that when the upper section is raised the two sections will separate, leaving an opening or port 12 ins. in height by the full width of the gate, thus permitting the water to fill the penstock and create a back-pressure. When the gate is balanced by this back-pressure the raising of the upper section is continued, and engaging the nuts on the rods connecting the two sections, the lower section is raised with it.

If it is required to keep the balance-port open while the gate is being lowered, two latch-rods are provided, running up to the gate-operating platform. By giving the latch-rods a quarter-turn the latches are thrown in and keep the sections separated. When the lower section is fully closed the latches are thrown out and the upper section is closed.

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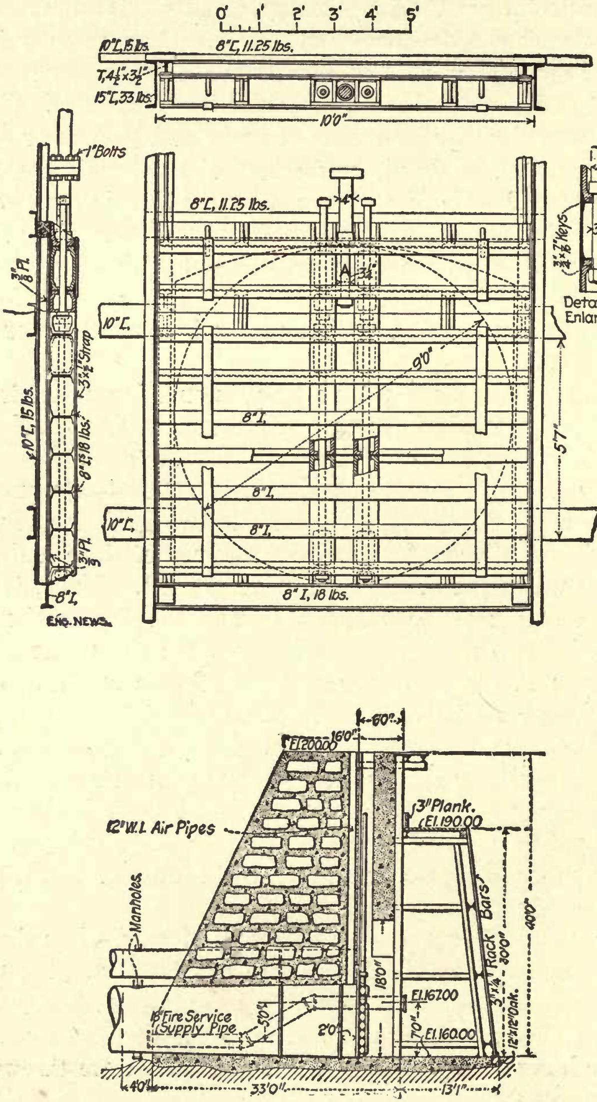
[1] See Frizell, "Water-power," p. 239.



FIGS. 72 to 75.—Steel Head-gate with Balance-port.

In Figs. 76 and 77 is shown a sliding steel head-gate provided with rollers to reduce the friction between the gate and the slides. The gate is kept tight at the sides by a curved plate, which is pressed against the plate fastened to the masonry, by the water-pressure.

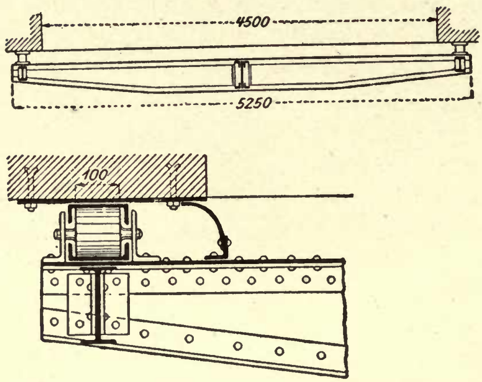
Figs. 78 and 79 represent a pivoted steel head-gate, which is also shown in position in the power-house cross-section Fig. 21. The pivot is so located that the gate is nearly balanced.

In cold climates all head-gates which raise above the water level should be so arranged that they are entirely below the surface ice when closed and entirely above the surface ice when open, to prevent them from freezing fast. In a well-known water-power plant in New York State, using very large sliding steel head-gates, it is often necessary in freezing weather, after the plant has been shut down over Sunday, to use dynamite and jack-screws on the penstock side of the gates, which is accessible, to get the gates started before they can be raised.

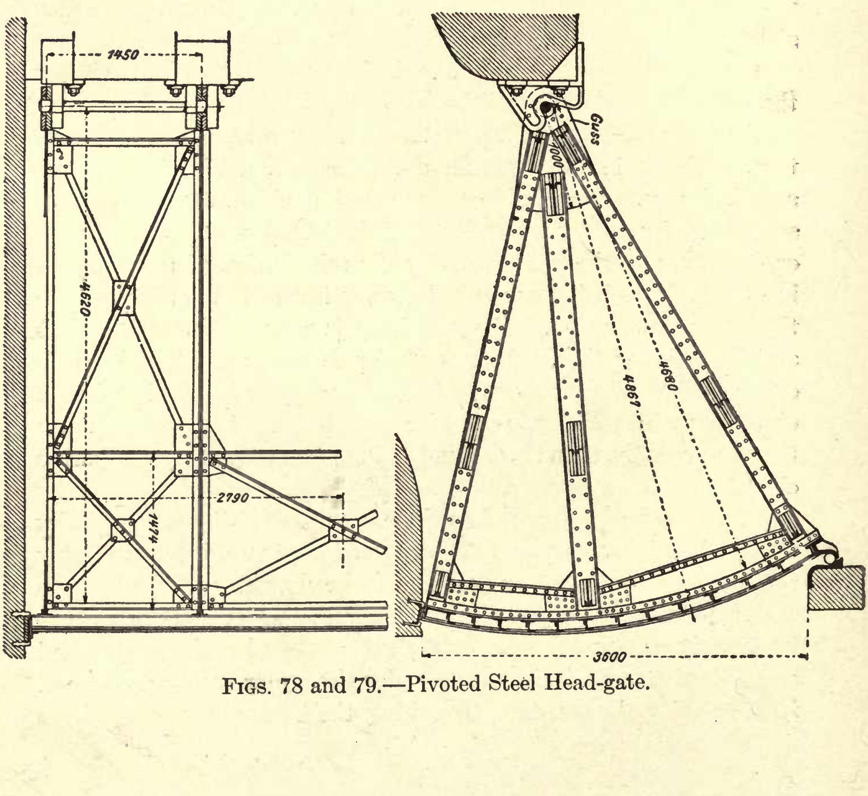
Gates sliding horizontally, and thus remaining always below the water level or surface ice, may be used in some cases.

An interesting head-gate, and deserving a more general application, is shown in the power-house cross-section Fig. 20. This is a cylinder gate made of cast iron and double-seated, the lower seat being formed by a ring fastened to the bottom of the headrace or forebay, and the upper seat by the raised edge of the dome or head, the dome being supported by vertical steel bars fastened to the lower seat or ring. The gate itself is a cylinder cast in halves and bolted together. This cylinder is connected to the ends of two yokes by four short links, and each yoke is attached to a chain, by which the gate-cylinder may be raised or lowered. In the center of the dome is a small filling-gate, moved by a separate chain.

The cylinder head-gate has the advantage that it is practically balanced and thus requires little power to operate, it requires only a movement equal to one quarter of the diameter of the inlet to open or close, it remains always below the water level or surface ice, and it can be made perfectly tight. Sand and stones rolled along the bottom of the headrace by the water can be prevented from entering the penstock by raising the lower seat or ring above



FIGS. 76 and 77.—Steel Head-gate with Friction-rollers.



FIGS. 78 and 79.—Pivoted Steel Head-gate.

the bottom. In localities where there is no ice formation a cylinder head-gate may be used, which is seated only at the lower end, while the upper end projects above the water level and is guided by rollers.

Stop-log slides should be provided in front of all head-gates, so that the water can be shut out and the gates and slides or seats made accessible for inspection and repairs.

Penstocks.—Penstocks or feeder-pipes should always be as short as possible, even when a shorter penstock involves a greater expenditure for excavation, etc. This is for the reason that the shorter the penstock the better it is for the speed regulation of the turbines, and the less steel-plate work has to be kept painted and repaired.

The following rules should be observed when determining the cross-sectional area of the conductors which convey the water to and from the turbines:

The speed of the water should be gradually increased from the speed in the headrace, usually 2 or 3 ft. per second, to the penstock speed, by means of a cone or taper piece. Near the lower end of the penstock the speed should again be gradually increased, so that the water will arrive at the guide-buckets with a speed equal to that with which it has to enter these guide-buckets. At the entrance of the draft-tube, or draft-tube elbow or tee, the water should have a speed equal to the absolute velocity with which it leaves the runner-buckets, and should then gradually decrease to a speed of about 2 or 3 ft. at the lower end of the draft-tube. A speed of 2 or 3 ft. is also usually chosen for the tailrace.

In general it should be stated: Avoid changes of speed of the water where possible, but where such changes are necessary make them gradually; also, avoid changes of direction of water, but where such changes are necessary use curves of long radius.

The arrangement often employed of having at the lower end of the penstock and at right angles to the same a drum or receiver of much larger diameter than the penstock itself, from which drum a number of turbines are supplied by branches set at right angles to the drum, must be condemned on account of the abrupt changes in speed and direction of the water.

All nozzles or branches of penstocks should be at an angle of

not over 45° to the penstock, or, in other words, the directions of flow of the water in the penstock and in the nozzle or branch should form an angle of not over 45° with each other. Directly beyond each nozzle or branch the diameter of the penstock should be reduced, to keep the speed of the water uniform.

When determining the speed for the water in the penstock all conditions should be carefully considered, and it should also be borne in mind that the friction loss in a penstock varies with the square of the speed.

Conditions making a low speed advisable are: Low head, large diameter of penstock, great length of penstock, many bends in penstock, variable loads on the turbines, regulation of speed of turbines by changing the amount of water used.

Conditions making a high speed permissible are: High heads, small diameter of penstock, short penstock, few or no bends in penstock, steady loads on the turbines, regulation of speed of turbines by by-pass.

Many hydraulic engineers employ in all cases a penstock speed of 3 ft. per second, but it is often of advantage to greatly exceed this velocity. From a great number of well-designed water-power plants constructed in America and Europe during recent years the writer has deduced the following table of highest permissible speeds of water in penstocks of a length of 1000 ft. or less, with easy bends, and provided with proper arrangements for the protection of the penstocks against water-hammer:

Diameter of penstock, in feet.	4	5	6	7	8	9	10	11	12
Speed of water, feet per second	12	11.5	11	10.5	10	9.5	9	8.5	8

In penstocks of 1 or 2 ft. diameter speeds as high as 20 to 30 ft. have been used. With very low heads the penstock speed is often limited by the amount of head that it is permissible to lose in the penstock.

The principal losses in the head of the water while entering the penstock and flowing through the penstock and draft-tube are due to the following causes:

(1) The entrance loss. This loss may be kept low by having

a large entrance connected to the penstock by an easy cone or taper piece. With the usual head-gate arrangement such large entrance openings require very heavy and cumbersome gates for penstocks of large diameter, but there is no reason why this taper piece could not be partly or wholly in front of the gate and inside the headrace or forebay. The penstock entrance should always be as much below the surface of the water as circumstances will permit.

(2) The friction loss: This loss may be kept down by a low speed of water, and by smooth interior of the penstock and draft-tube.

(3) The loss due to changes in direction of flow. This loss may be kept down by using as few and as easy bends as possible.

(4) The loss caused by changes in speed of the water. This loss is due to the conversion of part of the energy in the water into another form, and may be kept low by having as few and as gradual changes as possible.

(5) The loss due to the speed of the water while leaving the lower end of the draft-tube. This loss is equal to the velocity head, corresponding to the speed with which the water leaves the draft-tube, and may be kept down by making this speed low.

Many engineers regard the velocity head corresponding to the speed of the water in the penstock as a total loss, but this is, of course, not the case.

Penstocks are usually made of tank-steel, but for penstocks carrying water at high speeds shell-plate steel should be used, having an ultimate strength of 55,000 to 65,000 lbs. per square inch and an elongation of 22% in 8 ins.

A factor of safety of 3 to 4 may be used for short penstocks of large diameter, carrying water at low speeds and supplying turbines with steady loads or regulated by means of a by-pass. A factor of safety of 4 to 6 may be adopted for long penstocks of small diameter, carrying water at high speeds and supplying turbines with variable loads regulated by changing the amount of water used.

The factors of safety here given are fully sufficient in all cases, as steel-plate penstocks very rarely fail by bursting, but are destroyed by pitting, that is the formation of rust-holes. Usually

a higher factor of safety is employed for the lower end than for the rest of the penstock, as the effect of water-hammer is the greatest at the lower end. Large penstocks, carrying water under a very low head, have often to be made of thicker plate than the water-pressure would require, to prevent them from flattening or collapsing by their own weight when empty.

Long penstocks, carrying water at high speed, should be provided with a safety-head, besides the usual devices for the protection of the penstock against water-hammer. For this purpose a cast-iron or angle-bar flange is riveted to the lower end of the penstock, to which flange the head, closing the lower end, is bolted. The flange-bolts should have a factor of safety of not more than about half the factor employed for the rest of the penstock. Between flange and head a packing of dry white pine should be used which, when water is admitted to the penstock, swells and makes a tight joint. Where the end cannot be used for this purpose, large nozzles may be riveted to the penstock, located as nearly as possible in the line of the water-hammer, and closed by heads secured as just described. The end of the penstock or the nozzles should be so situated that, should the heads blow out, no damage will be done by the jet of water issuing from the opening. This arrangement will not only save the penstock and turbines from being wrecked in case of severe water-hammer, but also the power-house from being demolished by the water set free.

Ample air-inlets should be provided at the upper end of the penstock, as shown in Fig. 75, as otherwise—should the safety-head by some chance give way, or the turbine-gates or turbine stop-valves, if such are employed, be opened, while the head-gate is closed but the penstock full of water—the penstock might collapse by the vacuum created in its interior. Care must be taken to prevent the water in the vents or air-inlets from freezing, as this would render them useless.

A penstock which is carried for a considerable distance at about the same elevation as that of its inlet and with so little slope as to be nearly horizontal, and then descends to the power-house on a steep grade, is liable to collapse when the turbine-gates are opened quickly, as the water in the inclined part has

the tendency to increase its speed more quickly than tne water in the horizontal part, and may thus break away from the latter and cause a vacuum in the penstock. An air-inlet valve will prevent this, but it is better to have a small compensating or equalizing reservoir at the junction of the horizontal and inclined part of the penstock. Such a reservoir may be built of steel plate, concrete, or masonry, and will not only prevent the collapse of the penstock from the cause above named, but will also greatly improve the regulation of the turbines and decrease the water-hammer in the penstock, acting, in fact, in the same manner as a stand-pipe.

Expansion-joints in penstocks are not so important as is often asserted, as most penstocks contain bends which permit of a limited movement, large enough to compensate for expansion and contraction, but in a straight penstock, rigidly held at each end, the strains due to changes in temperature are very heavy. The amount of these strains depends on the modulus of elasticity and the coefficient of expansion of the material; being for medium steel equal to 200 lbs. per square inch for a change in temperature of 1° F. For example, a straight steel penstock 9 ft. in diameter and made of $\frac{5}{8}$-in. plate has a cross-section of metal, including the laps, of about 220 sq. ins., and if held rigidly at both ends will exert a thrust or pull of 44,000 lbs. or 22 tons for each degree of rise or fall in temperature; and assuming a rise and fall of 50° from a mean temperature of, say, 40°, or a total range of from $-10°$ to $+90°$, and further assuming that the penstock was erected at the mean temperature of 40°, the greatest thrust or pull will be 10,000 lbs. per square in., or 1100 tons for the whole penstock. These figures show that an expansion-joint should be provided in a straight penstock.

The lower end of a penstock should be held very securely in all cases to prevent forces due to temperature changes and other causes from throwing the turbines out of alinement, cracking the power-house walls, etc.

Steel-plate penstocks are usually made in small and large courses and lap-riveted. Butt-strap joints, with a single butt-strap on the outside, offer less frictional resistance to the flow of the water, but are more expensive. A manhole should be

provided at the upper end of a penstock, as shown in Fig. 75, and at the lower end also, if required. When repainting the inside of a penstock or repairing the same, the water leaking through the head-gate should be prevented from running down the penstock, and for this purpose a small outlet-nozzle, about 6 in. in diameter and closed by a blank flange, is provided at the lower side of the upper end of the penstock, as shown in Fig. 75, and by building a small dam of clay in the penstock, just beyond this nozzle, the leakage is prevented from flowing down the penstock. All openings in penstocks for large nozzles, branches, manholes, etc., should be reinforced by steel-plate rings, riveted around the openings, to make up for the material cut away by the opening.

If a stop-valve is to be used at the lower end of a penstock or its branch, a union similar to the one seen in the draft-tube of the turbine shown in Fig. 34, but with flanged and bolted joints, should be employed to facilitate getting the valve out for repairs and returning it into position, without moving the penstock or turbine-case.

As the exact length of a long penstock cannot be obtained beforehand, requiring perhaps surveys over rocky mountain-sides or through dense forests, a large course, called a shearing-strip, should be left out at about the middle of each long tangent or straight part. The provisional length of this shearing-strip should be about one half the length of a regular course, thus leaving one half the length of a course either way, to make up any inaccuracies in the first measurements for the penstocks. The curves of a penstock are usually built first and then the straight parts constructed, the shearing-strip being supplied from actual measurement, after all the rest of the penstock has been riveted up.

Penstocks should be calked both inside and outside, and the plates thoroughly cleaned by scrapers and wire brushes before painting. Of paints used for the protection of penstocks may be named the iron and lead oxide paints, the graphite paints, coal-tar, asphaltum, and the various patented compounds, but none can be said to satisfy all requirements.

The riveting, calking, and painting inside of a small penstock, where the workman has to lie down, or in a large penstock, where

scaffolds have to be used, and such penstocks running down a steep mountain-side, is a very arduous task and the workmanship should therefore be carefully inspected, as the men are liable to slight such work. In hot weather a penstock exposed to the sun's rays will become intolerably hot, and men having to work inside of such a penstock should have their working hours from about 10 P. M., when the penstock has had time to cool off, to about 9 A.M.

Masonry piers are often damaged by the expansion and contraction of the penstock they support, and the paint is rubbed off the penstock where it rests on the piers. Such unprotected places are hidden from view by the masonry, and are apt to corrode very quickly, as water is always retained between the surfaces of contact of the masonry and the penstock. It is therefore preferable to use steel piers on concrete or masonry bases, as shown in Fig. 80.

Such steel piers are cheaper than concrete or masonry piers; they leave every part of the penstock accessible for painting and repairs, and are free to swing on their bases, like inverted pendulums, to accommodate themselves to any movements of the penstock caused by changes in temperature. The uprights or posts of these piers are provided with bolt-holes, to fasten to them the studs for a housing over and around the penstock when desired.

Except where the distance between the penstock and the ground varies considerably, the steel piers are all made the same, and the variations in the height of the penstock above the rock or solid ground are made up in the height of the concrete or masonry bases. The uprights of the steel piers are anchored to the bases, or, if the latter are of small height, through the bases to the rock below.

A penstock running down a steep mountain-side must be prevented from sliding down the slope. Where concrete or masonry piers are employed, it is often sufficient to rivet short pieces of heavy angle-bars to the penstock and having these bars bear against the up-hill side of the piers, but with steel piers the penstock must be anchored to the rock or special anchor-piers. It is well to have, in any case, a specially heavy concrete or masonry pier at the lower end of the penstock, to prevent the latter from throwing the turbines out of alinement.

In a climate like that of the northern part of the United States and of Canada penstocks must be covered or boxed in to protect them from the extreme cold, otherwise ice will form on their inner

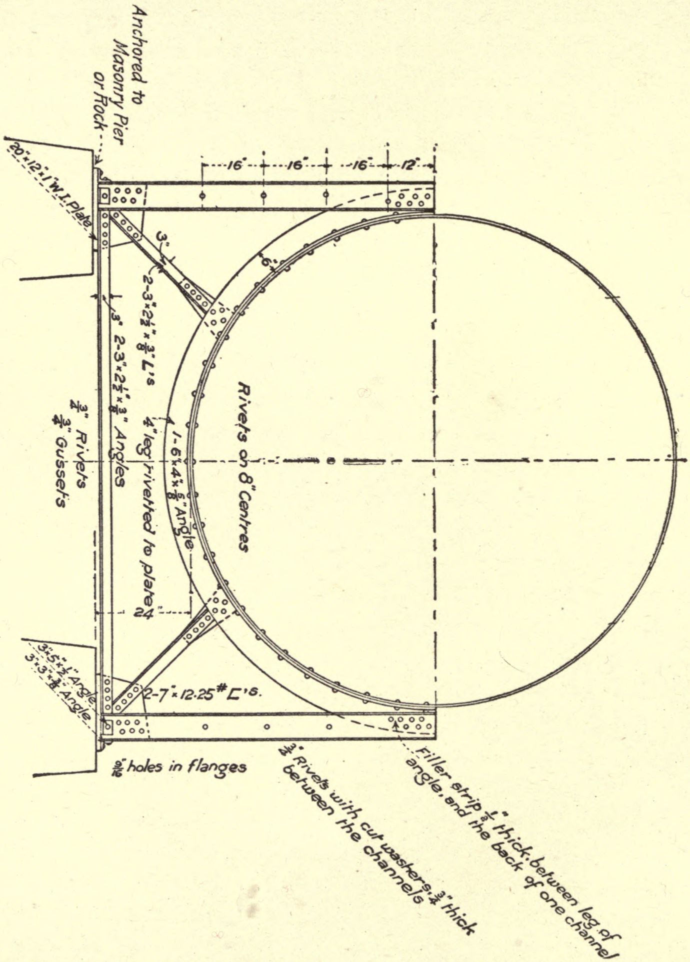


FIG. 80.—Steel Pier for Penstock of 9 Feet Diameter.

surfaces. At Grand'Mère, Que., a penstock of 14 ft. diameter, left unprotected during the first winter, was found to have its interior surface covered with solid crystal ice of from 12 to 18 ins. in thickness.

During midsummer the heat of the sun's rays, acting on an

empty penstock, will often injure the paint, cause it to blister off, and perhaps overstrain the penstock itself, and a covering will, therefore, prove an advantage both in cold and hot weather. Even in a well-protected penstock ice will be formed in severe weather when the water in it is allowed to remain stationary for more than a few hours at a time.

Where the ground under a penstock consists of earth it is preferable to bury the penstock below the frost-line, like the water-mains in a city street.

A buried penstock is free from the bending strains occasioned in a penstock supported on piers by the unsupported length between the piers, but a buried penstock of large diameter will require stiffening angles to be riveted to the upper half of its circumference, to prevent it from collapsing by the weight of the earth above it.

The cost of burying a penstock will be about the same as when masonry piers are used, as the following figures will show: A penstock 4 ft. in diameter, supported by masonry piers spaced 15 ft. center to center, requires piers of, say, $8 \times 6 \times 2$ ft. in mean dimensions, containing about $3\frac{1}{2}$ cu. yds. of masonry each, and costing, including the necessary excavation for the base, at least \$25 to \$35 a pier. A ditch 6 ft. wide and 9 ft. deep, in a length of 15 ft., contains 30 cu. yds., and, at 50 cts. per cubic yard for excavation and 30 cts. for filling and tamping, costs \$24, to which may be added \$6 for drain, lumber required, etc., making a price of \$30 per length of 15 ft. Further, while the penstock buried in the ground has the best possible protection against extremes in temperature, an additional expense will be required to protect the penstock supported on piers.

Under the penstock, in the center of the ditch, should be a drainage-ditch about 1 ft. square in cross-section, and filled with pebbles or broken stone, as used for concrete-making. The penstock should rest on short wooden blocks, and the main ditch should be left open during the first year, or for one winter season at least, after which the penstock is carefully inspected, recalked where leaky, and repainted inside and outside, after which the earth is packed under and around the penstock and the ditch filled in, removing the wooden blocks as the work proceeds.

Wooden penstocks or stave-pipes deserve a wider application than they have so far found in the Eastern States. Wooden penstocks are cheaper and will last longer than steel penstocks, need less protection against extremes in temperature, and require no painting. Their interior surfaces are smoother than those of steel penstocks and therefore offer less frictional resistance to the flow of the water.[1]

Wooden penstocks are made of staves from 2 to 4 ins. thick and from 6 to 8 ins. wide, planed to the proper shape and held together by round iron or steel rods, connected by hoop-locks. The staves must be thick enough or the hoops spaced closely enough to prevent the staves from bulging out between the hoops. Thick staves are usually provided on one edge with a bead of from $\frac{1}{16}$ to $\frac{1}{8}$ in. in height by $\frac{1}{2}$ to $\frac{3}{4}$ in. in width and located next to the inner side of the stave, as with such a bead it will require less strain in the hoops to make the penstock water-tight.

The joints at the ends of the staves are usually made by steel tongues, driven into kerfs. These joints must be well broken.

Curves in wooden penstocks require a long radius and therefore their horizontal and vertical alinement must be located on the ground, like a railroad line. The minimum radius, in feet, that can be used in a wooden penstock is about $R=12.5\times D_p\times t_s$, in which D_p is the inside diameter of the penstock in feet and t_s the thickness of the staves in inches. Where a smaller radius is required, a section of steel penstock has to be inserted in the wooden one for the purpose.

The wood employed should be clear and sound and free from pitch, so that the staves will become saturated by the water. The wood used for such stave-pipes is, in the order of its value for the purpose: California redwood, Douglas spruce (also called Douglas fir), spruce, white pine, southern pine, and cypress.

The staves of a wooden penstock that is not left empty long enough to allow the wood to dry will last much longer than the hoops, and the hoops may be renewed, when destroyed or weakened

[1] For a very complete paper on wooden penstocks see Mr. Arthur L. Adams, "Wood-stave Pipe: its Economic Design and Use," read before the meeting of the Am. Soc. C. E., Oct. 19, 1898; also an abstract in Engineering News, Oct. 27, 1898, p. 259.

by rust, by placing new ones between the old hoops, if the soundness of the staves will warrant it. A stop-valve should be used at the lower end of a wooden penstock and no head-gate at the upper end, to insure the penstock being always full of water, which may be shut out by the use of stop-logs in case of necessity. A wooden penstock buried in the ground may be left empty for some time, without danger of the staves drying out.

For heads of 200 ft. and more in height wooden penstocks are not economical, as the hoops require as much metal as the plates for a steel penstock.

Penstocks constructed of concrete and steel also deserve a wide application and should outlast both the steel and wooden penstock, as the steel rods are protected by the concrete.[1]

Instead of welding together the ends of the embedded hoops, these ends may be run past each other for a distance of from 30 to 40 times the diameter of the hoop-rod, or an inch or so of each end of the hoop-rod may be bent back flat on itself, and the ends run past each other for a distance of from 20 to 30 times the diameter of the hoop-rod. For small concrete penstocks steel wire wound spirally can be used to form the hoops.

For heads of 200 ft. and more in height penstocks built of concrete and steel are not economical, as the hoops require as much metal as the plates for a steel penstock.

Stand-pipes may be built either of steel plate or of concrete and steel. An excellent arrangement is to have a concrete base straddling the penstock, and the stand-pipe placed on top of this base, like a steel chimney or stack.[2]

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[1] An illustrated description of a ferro-concrete penstock will be found in Engineering News, Jan. 22, 1903, p. 74.

[2] For an illustration of such a stand-pipe see Engineering Magazine, Feb. 1903, p. 686.

CHAPTER IX.

THE DEVELOPMENT.

Developing a Water-power.[1]—The hydraulic engineer, before undertaking the development of a water-power, should make a thorough study of every aspect of the proposition, although this is often difficult, as promotors are generally not willing to invest money in the preliminary work until the capital has been procured and construction work is about to begin, yet the absence of a thorough knowledge of all the features involved in a development will often necessitate a change of plans during construction, abandonment of part of the work done, delay in completing the plant, extra payments to the contractor, or lawsuits.

The principal and often most difficult point to be determined is the discharge or rate of flow of the river. Except under very favorable conditions, a development will only pay if the whole plant can be run at all stages of the water and therefore the volume of river discharge, on which the development is based, should be as a rule the minimum low-water discharge of the average year, although for rivers having a widely varying minimum flow in different years the volume should be taken at or near the record minimum discharge.

Next in importance is the head, and it is essential to determine not only the head at low-water discharge, but also at times of flood-water discharge, as during floods the tailwater may be backed up to a height of 20 ft. or more.

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[1] It is not intended here to deal with the physical aspect of the development of a water-power or the design of dams, as these subjects are fully treated in such books as Merriman, "Treatise on Hydraulics"; Frizell, "Water-power"; Wilson, "Irrigation Engineering"; Wegmann, "Design and Construction of Dams"; Baker, "Masonry Construction"; etc.

The hydraulic engineer is often required to base his preliminary work on measurements furnished by some unknown engineer, but such data should only be used with the greatest caution. In his own practice the writer found some of the figures furnished for the river discharge to be from five to nine times the actual low-water discharge, and the head given to exceed the actual head by 25 to 75%. The engineer, when basing the preliminary work on such measurements, which he has no chance to verify, should save his own reputation by stating in his report on the proposition: According to the measurements furnished by, etc.

It will sometimes be found advisable, for economical reasons, to develop only part of the available head, as the utilization of the remaining head would increase the cost of the plant per horsepower to a disproportionate extent. In general it can be stated that the higher the head, the longer may be the water-conductors that can be used economically.

Before making any definite plans the engineer should, if possible, visit the site of the proposed development during a spring flood, to study the action of the water, ice, etc., and design his dam and head-works accordingly. The informations regarding such action of the spring flood, given by inhabitants of the locality, should be accepted only with caution.

The engineer is also sometimes furnished with a map showing elevations of the underlying rock, the soundings being made by ramming down a gas-pipe, but such soundings are very unreliable, as the pipe will often strike large boulders embedded in the earth, and thus show an apparent elevation of the rock which may be considerably above the actual one.

When calculating the stability of a bulkhead or dam where the penstock-inlets are located, the loss of weight in the bulkhead due to the displacement of concrete or masonry by the inlets must be taken into consideration; also, when calculating such a bulkhead for crushing and shear, the reduction in the horizontal crushing and shearing area, at the elevation of the center of the inlets, must be considered.

The modern hydraulic engineer should be an expert in concrete and concrete-and-steel construction.

The sand used in mortar- and concrete-making should be fre-

quently examined, as with the horizontal and vertical extension of the sand-pit its quality may change quite suddenly. Sawdust, discharged into a river, will usually cause the beach-sand below to be intermingled with fine sawdust of the same color as the sand. Such sawdust, of course, makes the sand unfit for mortar- or concrete-making, but it often requires a close examination to detect it. Wood-pulp refuse, discharged into a river, will also frequently make the beach-sand below unfit for use and cover the beach-pebbles with fine films of pulp, but by washing them thoroughly such pebbles may be employed in concrete-making.

To prevent tunnels and chambers in the concrete or masonry foundation of a power-house from being damp, such tunnels and chambers should be well ventilated.

The resident engineer should always endeavor to be on friendly terms with the contractor. If the contractor meets with any difficulties in his work, the engineer should do all in his power to aid him with his experience and knowledge, as the engineer is likely to meet with difficulties on the next day and the contractor's experience may prove useful to him.

For preliminary work the power may usually be calculated by assuming the total efficiency of the development, and with good judgment the result should be within 5% of the actual power. The total efficiency may vary from about 60% for developments with very long water-conductors, low-class turbines, etc., to about 85% for developments with open turbine-chambers, high-class turbines, etc.

If H_t is the total head utilized, Q the number of cubic feet of water per second, η_t the total efficiency of the development, and 62.3 the weight in pounds of a cubic foot of water at approximately 70° F., then the horse-power will be

$$\text{H.P.} = \frac{62.3 \times \eta_t}{550} \times H_t \times Q.$$

If the first term of the right-hand side of the equation, or $\frac{62.3 \times \eta_t}{550}$,

equals f, and its reciprocal, $\frac{1}{f}$, equals r, for the assumed total efficiency, then the horse-power will be

$$\text{H.P.} = f \times H_t \times Q = \frac{H_t \times Q}{r}.$$

VALUES OF f AND r FOR DIFFERENT EFFICIENCIES.

Total efficiency, η_t:	60%	65%	70%	75%	80%	85%
f:	0.068	0.074	0.080	0.085	0.091	0.096
r:	14.71	13.58	12.61	11.77	11.03	10.39

All important water-power plants, using large turbine units, should be arranged to have a separate head-gate and penstock and a separate tailrace, up to the point where the tailrace emerges from under the power-house, for each unit.

The head to be considered in connection with turbines is, of course, the head available or effective at the turbines, and, as a rule, turbines are chosen to give their best efficiency with the effective head obtained during low water, but sometimes they are selected for the normal or average head.

If a turbine develops a power E, under a head of H ft., uses Q cubic feet of water per second and runs at n revolutions per minute, then the same turbine will, under any other head H', develop a power E', use Q' cubic feet of water per second and run at n' revolutions per minute, or

$$E' = E \times \sqrt{\left(\frac{H'}{H}\right)^3}, \quad Q' = Q \times \sqrt{\frac{H'}{H}}, \quad n' = n \times \sqrt{\frac{H'}{H}}.$$

This is provided the head H' does not differ so much from the head H that the turbine will give different efficiencies under the two heads.

Engineers often appear to think that it is a special merit to install turbines of very large power, but the horse-power of a turbine should always be in proportion to the capacity of the plant. Thus in a plant of 10,000 H.P. capacity it would be wrong to install two 5000-H.P. turbines, as a breakdown would reduce the plant to one half of its capacity, while with four 2500-H.P. turbines a breakdown would reduce the capacity of the plant by only 25%.

In Europe all important plants which sell power have a spare turbine unit, always kept ready to set to work should any other unit break down, and it would be well if this plan were generally followed.

All turbines must have a capacity somewhat in excess of the maximum load, to have a margin for speed regulation. Reaction turbines usually give their best efficiency when discharging 80% of the amount of water discharged with full gate opening, and it is advisable to install turbines to develop the ordinary maximum of power required with this discharge, and to leave the remaining 20% of discharge for emergency loads and as a margin for speed regulation.

In a large plant, containing many turbine units, it is not necessary for the turbines to have a high efficiency with part gate, as the engineer in charge of the plant can attend to the larger changes in the demand for power by the starting and stopping of turbine units, while one turbine unit takes care of all minor fluctuations in the power required.

Some engineers think to be very far-sighted to specify dynamos to be capable of running for hours with an overload of 50%, and thus the turbine necessary to utilize such an overload capacity is required to run with only two-thirds of its full load and the corresponding decreased efficiency, during the whole time that the dynamo runs with its normal load. As large dynamos are usually designed to be capable of running from two to three hours with an overload of 25%, the dynamo with a 50% overload capacity is, as a rule, simply a larger-sized dynamo, sold under the name of a smaller-sized one, to suit the demand of the engineer. The better plan is, of course, to have an extra turbine and dynamo unit to take care of an exceptional or emergency load.

As already stated, special reaction turbines can be built which give their maximum efficiency at any desired gate-opening and corresponding discharge, but such turbines do not give as high a maximum and average efficiency as the ordinary turbine. Turbine specifications should not only state the power which the turbine is to develop with full gate-opening, but also the maximum power which will be accepted, as turbine-builders, to obtain the desired number of revolutions, often furnish a larger turbine than is other-

wise necessary, which means that the turbine has to run with part-gate and part-gate efficiency, even when developing the maximum power required. The amount of power to be accepted in excess, over the required maximum power, should be, under ordinary conditions, about 20% for a turbine of 100 H.P. and decreasing to about 5% for a turbine of 5000 H.P.

In the plans accompanying turbine specifications only such dimensions should be given as must be adhered to for some reason or other, leaving the turbine-manufacturer free to choose the dimensions that will best suit his designs or patterns.

The efficiency tests of a turbine should always be made after the turbine has been installed in the plant in which it is to run, and under normal working conditions. The turbine specifications should therefore state that the efficiency tests are to be made after the turbine is installed, and how the water and the power are to be measured. If the water is to be measured by a weir, as is usually the case, a drawing of the weir, giving all important details and dimensions, should accompany the turbine specifications, to avoid all disputes in this respect. As friction-brakes, especially for large powers, are rather expensive, the turbine-builder should have on hand different sizes of such brakes, and the turbine contract should include provisions for the loan of the necessary brake.[1]

Where the turbine is used for generating electric current, it is best to measure the power developed by means of the dynamo output, but, as some turbine manufacturers object to this, it should in such cases always be stated in the turbine specifications that the power is to be measured from the electric output. Of course, if the power of a turbine is to be measured in this manner, then the dynamo has to be properly tested for efficiency before leaving the shops of the builders, and this is usually done as follows:

The armature- and field-windings of a dynamo are tested for resistance and insulation, both while the dynamo is being built and after it is completed, and the dynamo is then tested for efficiency by running it at full speed and with various outputs. The power for driving the dynamo is usually furnished by a motor

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[1] An illustrated description of a hydraulic friction-brake for large powers will be found in Engineering News, May 19, 1904, p. 474.

of known efficiency, and whenever possible the output of the dynamo under test is in turn used to furnish the greater part of the current required by the motor. If two or more dynamos of equal size are being built for the same plant and at the same time, the running tests are made by setting two of the dynamos with their shafts in line, their coupling ends together, and coupling them. One of the dynamos is then used as a motor and the other as a dynamo, and the difference between input and output, that is the amount of electric power or watts required from some outside source, is the power loss of the combined motor and generator, and one half of it is assigned to each dynamo. Of course this is not quite correct, as the efficiency of the different dynamos may not be the same, but the difference in efficiency for large generators will rarely exceed $\frac{1}{2}\%$.

While testing a turbine, the output of the dynamo is usually absorbed by a rheostat. If the power is very large or no other rheostat is on hand, a water-rheostat is constructed. Such a rheostat consists of two flat iron or steel plates, one laid horizontally on the bottom of the river or a pond, and the other supported horizontally above it in the water, in such a way that it may be raised or lowered. A short circuit would be caused if the upper plate should accidentally drop on to the lower one, and to prevent this two scantlings or small timbers are laid on top of the lower plate.

Each plate is connected to one of the poles of the dynamo. For alternating-current dynamos, one set of plates is required for each phase. By raising or lowering the upper plate, that is by increasing or decreasing the distance between the plates, the output in amperes of the dynamo is decreased or increased respectively. About 50 amperes should be allowed per square foot of each plate, and the size of the plates chosen accordingly. If the water is very pure, the rheostat should be constructed in a wooden tank or pond, or a pit dug for the purpose, and salt added to the water to increase its conductivity

The power output of the dynamo is measured by a special test watt meter, two such meters being required for polyphase dynamos. The watt meter or meters should be loaned from the builders of the dynamo, and the dynamo contract should

therefore include provisions for the loan of the required test watt meter or meters.

Surface, Anchor, and Frazil Ice.—With the exception of some of the Southwestern States, the greatest difference between high- and low-water discharge of North American rivers is to be found in the northern parts of the United States and in Canada, that is in localities having a long and severe winter, the reason being that, while there is usually an abundant amount of rainfall during the summer, nearly the whole of the precipitation during the winter is in the form of snow, of which by far the greater part accumulates until the warm weather sets in, when the whole accumulation runs off in a comparatively short time, thus forming the spring floods.

Northern rivers, like those located farther south, have a period of low water during the dry season, that is in August or the first half of September, but the lowest water of the year occurs usually during the latter half of February and the first half of March, being due to the accumulation of the precipitation and the retaining of part of the water in rivers and lakes in the form of ice.

For rivers having a drainage area of 500 square miles and over, the average discharge during the period of winter low water may be considered to be one half of a cubic foot per second per square mile of drainage area, but often is less in certain rivers or in winters of great severity, as will be seen from the figures given here for three rivers of the Province of Quebec, the discharge being measured with current-meters through holes cut into the ice.

St. Maurice River: flowing north to south; drainage area 18,000 square miles, almost entirely covered by forest; winter low-water discharge 7515 cu. ft., or 0.418 cu. ft. per second per square mile.

Chaudière River: flowing east to west; drainage area 2600 square miles, almost entirely denuded of forest; winter low-water discharge 620 cu. ft., or 0.24 cu. ft. per second per square mile.

Metabetchouan River: flowing south to north; drainage area 890 square miles, entirely covered by forest; winter low-water discharge 380 cu. ft., or 0.43 cu. ft. per second per square mile.

On account of the floating ice it is almost impossible to make even approximate measurements of the river discharge during

the spring high water, except where an overflow-dam exists, as is the case on the Chaudière River. From the records kept by the owner of this dam, the spring high-water discharge in certain years has been as much as 121,000 cu. ft., or 46.54 cu. ft. per second per square mile, that is 195 or practically 200 times the amount of low-water discharge. In this case the great difference between high- and low-water discharge is partly due to the fact stated above, that the drainage area is almost denuded of forest. The influence of the forest during the spring high-water season consists chiefly in the retardation of the thawing of the snow and ice. In the forests snow may still be lying to a depth of 3 ft., while in the open country it has almost entirely disappeared.

The period of winter low water is, of course, very unfavorable for water-power electric plants supplying light, as the lowest water occurs at a time when the demand for light is nearly at its maximum.

In cold climates the greatest care and judgment are required in designing a water-power development, as the pressure of the ice, both of the sheet as formed and of accumulations of floating ice-floes, is often tremendous, and the chief aim of the engineer should always be to prevent the ice from reaching and exerting its pressure upon the structures of a development. While the stationary sheet or surface ice rarely gives trouble, the accumulations of ice-floes often become very dangerous.

Where the ice goes out at or near the highest stage of the spring flood-water, as is mostly the case, the conditions are the most favorable. Where the ice or part of it goes out after the spring flood, danger can be averted in most instances; but where the ice or a large part of it comes down the river before the spring high water has arrived, conditions usually are dangerous.

Most head-works of a development obstruct the river channel to some extent, especially a dam across the river; and with the ice arriving without sufficient water to carry it around or over such obstructions, accumulations will be formed so fast that even a large ice-sluice and the liberal use of dynamite cannot prevent a serious ice-jam. When high water sets in and finds the channel blocked by ice, it rises rapidly, flooding the surrounding country, until the hydraulic head has increased sufficiently to produce an ice-shove, that is a down-stream movement of the whole accumu-

lation in a body, carrying before it guide-booms and cribs and perhaps the whole head-works of the development.

The direction of the wind is often a very important factor in the formation of an ice-jam.

When an overflow-dam is to be built across a river in which ice-jams occur, a dam section having a vertical up-stream face should be avoided. The triangular cross-section, so frequently used for timber dams, having both up- and down-stream face inclined to form an angle of about 30° with the horizontal, is perhaps the best that can be chosen, as with such a section the ice cannot exert pressure against the face, but will be forced up the incline and over the crest and then shoot down the lower face and away from the toe.

Dams with a vertical or nearly vertical up-stream face should be provided with an apron starting from the top of the up-stream face and sloping downward at an angle of about 30°. Such aprons are usually built of timber, with only the upper part covered by planks, and serve the same purpose as the inclined up-stream face of a timber dam.

The down-stream toe of overflow-dams is in all cases to be well protected from the impact of the ice coming over the dam; especially masonry dams with ogee-shaped down-stream face are very liable to have the lower part of the ogee curve knocked off.

High overflow-dams should be avoided where possible, and a low dam and penstock used instead, even if the first cost is greater and a close speed regulation of the turbines made more difficult.

Earth dams or embankments should be kept out of reach of the floating ice, as large floes will cut through any slope-paving and thus may cause the destruction of an earth dam.

Surface ice on fast-flowing rivers is formed as bordage ice, that is by starting along the shore and extending outward. Surface ice on stationary bodies of water, such as lakes, etc., or on slowly flowing rivers, begins with the formation of needle-crystals all over the surface which, increasing in number and size and freezing together, form the surface sheet. Surface ice grows by being cooled below the freezing-point and abstracting latent heat from

the water in contact with its lower side, which causes such water to freeze to the surface ice. The regular surface ice will rarely grow to a thickness of over 3 or 4 ft., but anchor-ice and frazil, where such is formed, is carried below the surface ice and, adhering in great quantities to the lower side of it, will increase its thickness by freezing to it. In this manner the surface ice may increase to 20 ft. in thickness, being often solid to the very river bottom. In the St. Lawrence River the anchor-ice and frazil at times form masses hanging from the surface ice to a depth of 80 or 90 ft. below the water level.

In the United States it is generally supposed that the terms anchor-ice and frazil are synonymous, and this mistake is even to be found in dictionaries; but anchor-ice and frazil are actually two entirely different formations. The Montreal flood commission, which may be regarded as an authority on the subject, says in its report, printed in 1890: "Frazil, as distinguished from anchor-ice, is formed over the whole unfrozen surface [of the St. Lawrence River] above and below Lachine Rapids, between Prescott and tide-water and wherever there is sufficient current or wind agitation to prevent the formation of bordage-ice."

As it is of importance to the hydraulic-power engineer to foresee what he may have to contend with in the way of ice formation in any particular locality, the writer has given here the causes and principles of the formation of anchor-ice and frazil.[1]

Anchor-ice is formed along the bottom of rivers, creeks, canals, and lakes of not over 30 to 40 ft. in depth, and is of granular and

[1] Mostly from reports on the investigations made under the direction of Prof. Callendar of McGill University, Montreal, Que. by Dr. H. T. Barnes, of McGill University. These investigations were carried on during the winter of 1895 to 1896 at a place opposite Montreal, taking the water temperatures through holes cut into the ice, and during the winter of 1896 to 1897 in the open water of the Lachine Rapids. The temperatures were measured by an electrical thermometer with platinum resistance, indicating differences of 0.0001 of a degree Centigrade, when used with laboratory facilities while in the field its accuracy could be relied on to 0.001° C. These reports will be found in part in the Trans. Can. Soc. C. E., vol. 15, 1901, pp. 78 to 95.

See also "Experience with Anchor-ice at the Detroit Water-works and Elsewhere," by C. W. Hubbell, in The Michigan Technic, University of Michigan, 1903; also in Engineering News, Aug. 13, 1903, p. 147.

porous texture and of little strength. The formation of anchor-ice is caused by the radiation of heat from the river or lake bottom into space, the radiation taking place through the water and the atmosphere and even through thin, clear ice. The river or lake bottom, cooled below the freezing-point and abstracting latent heat from the water in contact with it, causes such water to freeze to the bottom. The radiation of heat through the ice was demonstrated by Professor Tyndall, who brought platinum to red heat by concentrating the sun's rays through a lens made of ice. After the anchor-ice has commenced to form, the heat from the river-bottom is radiated through the anchor-ice or conducted through it and radiated from the surface of the anchor-ice, causing the anchor-ice to grow in thickness.

Conditions favorable to the formation of anchor-ice therefore are a clear cold night, shallow open water or shallow water only covered with a thin clear ice-sheet. The radiation of heat from the river bottom into space, and with it the formation of anchor-ice, is partly or wholly prevented by cloudy weather, thick, rough or granular surface ice and heavy snow on the surface ice, while bright sunshine, striking the river bottom, counteracts the radiation of heat and thus also prevents the formation of anchor-ice. When anchor-ice grows thick it will, owing to its granular texture, retard or entirely prevent the radiation or conduction of heat through it, and further heat, being conducted from the interior of the earth to the surface of the river bottom, will melt off the hold of the anchor-ice. This melting off will also occur when the sun's rays penetrate through the water to the anchor-ice, thus causing the anchor-ice to appear on the surface of the water when the sun is high and warm. During the investigations it was found that the thermometer while in the water would show a higher temperature than the water itself when struck by the sun's rays.

In lakes and gently flowing rivers the warmer water, owing to its specific gravity, which is greatest at 39° F., sinks to the bottom, and no anchor-ice can therefore be formed until the whole body of the water has been uniformly cooled to the freezing-point.

However the amount of anchor-ice formed by radiation from the river bottom is only small, but this amount may be rapidly

increased by frazil adhering and freezing to the anchor-ice, often producing branching or tree-shaped formations of ice, which grow further by entangling surface-formed ice carried down by the current.

The formation of frazil or needle-ice takes place in rapids, where the velocity or agitation of the water prevents the formation of surface ice. With the temperature of the air much below the freezing-point, the churning water becomes slightly undercooled, that is it falls to a temperature not exceeding 0.01° C. below the freezing-point. This undercooling extends to the depth to which the water is agitated, and throughout this depth frazil is formed in needle-shaped crystals.

These crystals are usually formed around minute particles of material suspended in the water, which grow colder than the water itself by radiation, and act as a nucleus or starting-point for the crystals.

The size of the frazil crystals is the larger the less the velocity or agitation of the water is and the more it is undercooled, varying from the size of a small darning-needle to the size of an ordinary lead pencil; also, the more the water is undercooled the greater is the amount of frazil formed, and this formation may be so great that the water will become like a thin paste and have a dull sandy color.

Frazil is very sticky while the water is undercooled, and will adhere and freeze to surface ice and to any other object it comes in contact with. However, the greatest trouble is caused by the frazil crystals freezing to each other and to floating anchor-ice, forming lumps or agglomerations of ice, which cannot be stopped by glance booms and are capable of choking anything from a turbine-bucket to a head- or tail-race.

Rapids above a water-power plant have the advantage that they will break up the large floes of surface ice, which might damage the dam or head-works of the development. but the choking of the water-conductors, often requiring part of the river-flow for scouring to keep these conductors clear, and the backing up of the tailwater caused by the accumulations of anchor-ice and frazil, at a time when the river-flow is at its lowest stage, are serious disadvantages. At one large Canadian water-power plant

the effective head is often reduced from 11 ft. to 6 ft. or less. At times 30% of the waterway of the St. Lawrence River is blocked by agglomerations of anchor-ice and frazil, and the amount of such ice in the river is as great as the amount of the water itself.

In water which is little agitated anchor-ice and frazil rises and freezes to the lower side of the surface ice.

The only feasible way in which to reduce or prevent the formation of anchor-ice and frazil is to drown out the rapids by the building of a dam, so that continuous surface ice will be formed for many miles above the water-power plant.

Measurement of Water for Selling Water-power.—Whereever the owners of a water-power are selling power in the form of water under a head, that is power-water, some means have to be employed to measure the amount of water taken by the user. No great refinement is required in these measurements, but the apparatus must be simple, so that the readings may be taken by a man of ordinary intelligence. Of the means employed for such measurement, the following have been used to a greater or less extent:

1. Turbine regulating-gates. The turbines to be used are tested in a testing-flume or in place after being installed, for the amount of water discharged with different gate-openings and with the normal head or, should the head be very variable, with different heads, between the possible minimum and maximum, under which the turbines are to be operated. A scale is attached to each turbine, showing the gate-opening, and this scale is read at regular intervals of a few hours by an employee of the owners of the power. With variable heads gage readings, giving the head, are taken at the same time.

The turbine regulating-gates are by far the most widely employed means for measuring power-water.

Where the power is used for generating electric current, recording wattmeters will at once show the output in electric power, and from these records and the previously determined efficiency of the turbines and dynamos the amount of power-water used during a day, week, or month can readily be ascertained.

2. Water-meter in by-pass. The meter is placed in a by-pass to reduce the size of the meter, to prevent stoppage of main pipe or

penstock, when the meter is clogged or rendered inoperative, and to reduce the great friction and consequent loss of head caused by the meter. This method is hardly advisable for measuring water in connection with water-powers for several reasons, namely: The ratio of meter-readings to actual flow in main pipe varies very much, according to the type of meter, velocity of flow in main pipe, and condition in which the meter is kept. No reliable figures are on hand for this ratio for large pipes. With a ratio of meter diameter to pipe diameter as 1:3, the error in measurement rises to 6.5%, which would be increased as this ratio is increased. A screen, to protect the meter from clogging, could be placed in such a way that all solid matter, as chips, bark, etc., will go past it, but grass and other fibrous matter will soon cover the screen with a felty layer. No direct reading can be made for a day's or week's flow, as the ratio of meter reading to actual flow in main pipe varies so much, being, for example, 2.1 for a flow of 2.7 cu. ft., and 2.8 for a flow of 84.5 cu. ft. per minute; therefore the flow can only be obtained by taking the movement of the meter for, say, one minute, from which the actual flow for this particular minute may be calculated. Each meter has to be carefully tested in connection with its main pipe for all speeds of water. In cold weather the meter has to be protected against freezing. A Venturi meter can be substituted for the ordinary water-meter, thus avoiding all moving parts. Frazil may choke the by-pass entirely.

3. Pitot gage. This meter consists of two tubes inserted into the penstock, one giving the static, the other the static plus the impact or velocity head, the difference of head indicating the velocity of the flow and thus the actual flow. Each tube is provided with a float, the float in the static pipe carrying the scale and the float in the impact pipe carrying a rod with an index. By arranging the scale properly, the velocity or volume of flow can be read at any moment, and a recording apparatus can be attached showing the mean and total flow for a day, week, or month. Velocities as low as 4 ins. per second are indicated, and the error is said not to exceed 3%. The pipes can be made large enough so they cannot be clogged by small floating matter. The pipes, however, reaching into the middle of the penstock, are an impediment to the water, and large pieces of wood, bark, etc., which may enter

the penstock through places where the rack-bars have bent or spread, may make the gage inoperative.[1]

In cold weather the tubes have to be protected against freezing and the lower ends are liable to be choked by frazil.

4. Venturi meter. This meter consists of two cones, jointed at their small end by a throat-piece. The difference of pressure between large cone ends and throats indicates the velocity and volume of flow at any moment, and a recording-apparatus can be attached showing the mean and total flow for a day, week, or month. The error is said not to exceed 2%. With this meter, however, the accuracy depends on the exact size of the throat, and this will be soon enlarged by the sand carried in the water. The small connections between air-chamber and throats are likely to be clogged. For large quantities of water the price of the meter is very high and the size is enormous. By reducing the size, to lessen the price of the meter, the loss of head is rapidly increased.

In cold weather the throat and air-chamber have to be protected against freezing. The connections between throat and air-chamber are liable to be choked by frazil.

5. Measuring-gate. The meter consists of a gate with opening set and kept in a fixed position to pass the amount of water contracted for. The opening must be under low-water level on the up-stream side. A float, protected by a box, is provided on both the up-stream and the discharge side of the gate, one carrying a scale, the other a rod with an index. By arranging the scale properly, the velocity or volume of flow can be read at any moment, and a recording-apparatus can be attached showing the mean and total flow for a day, week, or month. Low velocities will be indicated and the error will not exceed 2%. With this meter, however, a correction has to be made when the gate-opening is submerged

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[1] For a paper on the Pitot gage see "Pitot Tubes; with Experimental Determinations of the Form and Velocity of Jets," by J. E Boyd and H. Judd, read before Section D of the Am. Assoc. for the Advancement of Science, Dec. 29, 1903; also reprinted in Engineering News, March 31, 1904, p. 318.

For a description of a Pitot gage with recording apparatus see "The Cole-Flad Photo-Pitometer and its Use in Studying Water Consumption and Waste," in Engineering News, Feb. 5, 1903, p. 130; also papers on the same apparatus, by Edw. S. Cole, in Trans. Am. Soc. C. E., vol. 47, p. 275 (1902); and Jour. Western Soc. C. E., vol. 7, p. 574 (1902).

on the discharge side, the flow with wholly submerged opening being, according to Weisbach, 98.67% of the flow when opening is wholly above water on discharge side. This is a reduction of flow equal to 67 H.P. in 5000 H.P. If the gate-opening is just large enough to pass the water contracted for, a compensating reservoir is required between gate and penstock entrance, for temporary overloads. This can be avoided by making the gate-opening large enough to pass all the water required at times of greatest possible overload. Where the locality is favorable, a measuring-weir in the tailrace may be used, instead of the gate above the turbines.

The float-boxes have to be prevented from freezing.

With all the above devices, the actual head can be taken into consideration when figuring the horse-power used.

All recording-devices have, of course, to be protected from the weather and extreme cold.

Cost of Water-power.—Whenever the development of a water-power for the purpose of selling water or mechanical or electrical energy is under consideration, the most important question to be decided is: What is the limit of cost per horse-power that may be expended for a development and still leave the plant a financial success, or what is a reasonable price to be charged per horse-power per year?

A great amount of data has been published in regard to the cost of hydraulic power and power-plants, but as water-powers present an infinite variety of conditions, such prices of other plants should only be used with the greatest precaution. A few general figures, intended to apply to conditions at present prevailing in the northern part of the United States and in Canada, may be given here.

A water-power electric plant, including transmission line and substation, where such are required, but without the local distribution, should not cost more than $100 per electrical horse-power if situated in a remote location or in a farming district; but $150 to $200 may be expended per electrical horse-power for power-plant, transmission, and substation, if the power can be sold in a large city or industrial district.

The price charged for power-water per gross horse-power per

year, delivered at or near the customer's turbines, may be taken at from \$5 to \$15, the lower figure being for remote locations, low heads, and large powers, and vice versa. The price of \$15 to \$25 per mechanical horse-power per year at the power-house, or of \$25 to \$50 per electrical horse-power per year delivered to the customer, may be taken as the limits paid at present. Here, again, the lower price is for remote locations and large powers, and vice versa.

It is also safe to state that in a climate such as that of the northern part of the United States and in Canada, with its long and severe winters, it does not pay to develop a water-power if the power produced will cost more than 75% of the amount for which steam-power could be produced in the same locality.

In Canada, with the great number of water-powers yet undeveloped or only partly utilized, it must be regarded as poor policy to install a larger plant than can be run at all stages of the water, or to have a great proportion of power dependent upon storage lakes during the months of low water.

A water-power requiring an auxiliary steam-plant during the low-water season can only pay if either the cost of the development is exceptionally low or the locality very favorable for the sale of power or both.

It may be noted here for comparison that in large steam-power electric plants in the Eastern and Central States the cost of the coal forms 60% of the total cost of the electric current.[1]

[1] According to Mr. H. G. Stott, in the discussion of M. P. Junkersfeld's paper "Multiple versus Independent Operation of Units and Central Stations," before the Am. Inst. E. E., April 24, 1903.

BRITISH AND METRIC MEASURES AND VALUES.

British and Metric Measures and Values.—Below are given the more important measures and values used in hydraulic-power engineering, according to both the British and the metric system, as some of the illustrations shown here contain metric dimensions and as the American engineering press sometimes prints accounts and illustrations of turbines or water-power developments taken from German or French contemporaries, and also for the convenience of those studying German or French works on turbines or hydraulic-power engineering or wish to look up the German books and periodicals referred to in the foot-notes.

While most text-books give the equivalents to more decimals, the writer has found the figures given below sufficiently accurate for all practical purposes.

When using metric formulæ for the design of machinery, the writer has found it very convenient to have a four-foot folding rule, divided to inches and sixteenths on one side and to centimeters and millimeters on the other. By holding the rule in the hand and placing the thumb-nail on the edge. in line with the dimension to be converted, the equivalent may at once be read on the other side of the rule. With rules having the subdivisions at opposite edges, a straight strip of writing-paper, folded across the two faces or around the rule, will enable one to read the equivalents in a similar manner. This method is sufficiently accurate to read to one thirty-second of an inch or to one millimeter.

1 foot = 0.305 meter. 1 inch = 25.4 millimeters.

1 meter = 3.28 feet = 39.37 inches = $39\frac{3}{8}$ inches.

1 mile = 5280 feet = 1.61 kilometers.

1 kilometer = 3281 feet = 0.621 mile.

1 square foot = 0.093 square meter. 1 square meter = 10.76 square feet.

1 square mile = 2.59 square kilometers. 1 square kilometer = 0.386 square mile.

1 cubic foot = 0.0283 cubic meter = 28.32 cubic decimeters.

1 cubic decimeter = 1 liter = 0.0353 cubic foot. 1 cubic meter = 35.3 cubic feet.

1 pound = 0.454 kilogram. 1 kilogram = 2.2 pounds.

1 ton of 2000 pounds = 0.9072 metric ton = 907.2 kilograms.

1 metric ton = 1000 kilograms = 2204.6 pounds.

Weight of 1 cubic foot of water at approximately 70° F. or 21° C. = 62.3 pounds. 1 cubic decimeter of water at approximately 21° C. = 0.998 kilogram.

Weight of distilled water at 4° C. = 39° F.

1 cubic decimeter = 1 kilogram. 1 cubic meter = 1 metric ton.

1 metric or mechanical atmosphere = 1 kilogram per square centimeter = 14.2 pounds per square inch = height of barometer 73.5 centimeters = 29 inches.

1 atmosphere = 1.0333 kilogram per square centimeter = 14.7 pounds per square inch = height of barometer 76 centimeters = 30 inches.

Theoretical height of suction at sea-level = 34 feet = 10.33 meters.

1 foot-pound = 0.1383 meter kilogram.

1 meter kilogram = 7.233 foot-pounds.

1 British horse-power = 550 foot-pounds per second = 1.014 metric horse-power.

1 metric horse-power = 75 meter-kilograms per second = 0.986 British horse-power.

Acceleration of gravity $= g = 32.16$ feet $= 9.81$ meters.
$\sqrt{2g} = 8.02$ for British and $= 4.43$ for metric measure.

1 cubic foot per square mile = 10.93 liters per square kilometer.
1 liter per square kilometer = 0.0915 cubic foot per square mile

Weight of bars, structural shapes, etc.
1 pound per foot = 1.488 kilograms per meter.
1 kilogram per meter = 0.672 pound per foot.

APPENDIX.

ELEMENTS OF DESIGN FAVORABLE TO SPEED REGULATION IN PLANTS DRIVEN BY WATER-POWER.

A paper presented at the 16th General Meeting of the American Institute of Electrical Engineers, Boston, June 27th, 1899.

By Allan V. Garratt.

In this paper the writer will endeavor to describe those peculiarities of design of plant which have a special bearing on speed regulation, but no attempt will be made to discuss the theory, mechanical construction, or merits of the various water-wheel governors on the market.

The engineer is often confronted with the problem of designing a plant upon an undeveloped or partly developed water-power, and the desired end is to come out with a plant of good mechanical and electrical design and yet have it such that the speed of the electrical apparatus may be maintained within comparatively close limits under any load variations which can possibly occur, and to maintain the speed within very close limits under any working-load variations.

The kind of generating apparatus used, and the nature of the load, predetermines the degree of regulation which must be obtained under both accidental and working conditions, but it is quite evident that the tendency in modern plants is in the direction of apparatus which requires closer speed regulation and more facility in handling the speed than heretofore.

It is quite possible to obtain on the market water-wheel gov-

ernors which will—provided the design of plant is good—give quite as good a speed regulation as could be obtained if the plant were driven by first-class steam-engines.

There is more than one water-driven electric plant in this country where auxiliary steam-plants are used in which the speed is fully as constant while the load is carried by the water-wheels as while it is carried by the steam-plant. The plants of the Derby Gas Co., the Pawtucket Electric Co., and the Woonsocket Electric Co. may be referred to as examples illustrating the above fact.

The largest accidental load variation which can occur is evidently an instantaneous change amounting to the full capacity of the water-wheels. The working-load variations may be anything less than this.

The writer has found, in a practice amounting to something over 90,000 H.P. of water-wheels in the last four years, that with good water-wheels properly set and rigged, and controlled by governors of suitable design, the speed may be held within 5 or 6% of normal upon circuit-breakers opening under full load, and that the speed may be brought back to normal in from five to fifteen seconds, depending upon the amount of kinetic energy in the rotative parts and moving water-column. With incandescent loads of the ordinary type, a recording tachometer will show a practically straight line. With ordinary electric-railway loads speed variations of about 3% as a maximum may be expected. These figures are not intended to be of universal application, but are for simply showing the present state of the art. It should here be added that governors can be obtained which will permit any number of independent water-wheel units driving electrical units connected in parallel to be operated with perfect convenience and safety. It should also be noted that in the case of alternating units it is perfectly easy to get them at speed and in step for multiple connection without undue delay, and without any hand regulation.

These desirable ends cannot, however, be obtained to their fullest extent if the general design of the hydraulic portion of the plant is bad. We will now consider those things, aside from the governor itself, which tend to make the regulation good or bad.

As a preliminary thought let us consider for a moment that

the problem is quite different from steam-engine governing, which naturally comes to the mind in this connection, for the reason that water is heavy, practically non-compressible or non-expansive, and must be transmitted to the water-wheel in large volume and at low velocity; while steam is light, highly compressible and expansive, and may be transmitted to the engine in small volume and at high velocity. From this it follows that the engine-valves are small, light, and may be perfectly balanced, while water-wheel gates are necessarily large, heavy, and are frequently—although often unnecessarily—out of balance. The inertia of the steam may be always neglected; the inertia of the water must be always considered.

The problem governing a water-wheel, then, involves moving large volumes of a heavy practically incompressible fluid acted on by the force of gravity alone, and of moving ponderous gates; and this must be done with absolute precision and great promptness. Also adequate provision must be made for the momentum and inertia of the moving water and mechanical parts.

To put our minds in a proper attitude to approach this subject let us refresh our memories in regard to some of the relations of force, mass, velocity, and time.

We have here a mass free to move. Its property of inertia prevents its moving until some force is applied to it. When, however, I apply for a moment the force of my hand it begins to move, and when I stop pushing it, it continues to move with a fixed velocity until I apply to it the same amount of force I used to put it in motion which brings it to a rest, and it cannot move again until a new force is applied to it.

I have here a pendulum beating seconds, and here (Fig. 1) are two masses consisting of pairs of balanced weights suspended by fine wires over pulleys which have as little friction as possible. One of these masses is twice as great as the other. If we apply to them equal forces in the shape of small additional weights, we will find that at the end of 1 second the smaller mass has acquired twice the velocity of the larger mass; or, in other words, where forces are equal the rates of acceleration are inversely proportional to the masses.

But now I have here two equal masses (Fig. 2), and if we apply

to them two forces—one twice as great as the other,—we will observe that at the end of 1 second the velocity of the mass acted upon by the larger force is twice as great as that of the mass acted upon by the smaller force; or, in other words, where masses are equal the velocities are proportional to the forces.

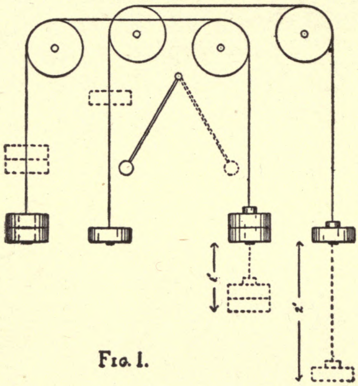

Fig. 1.

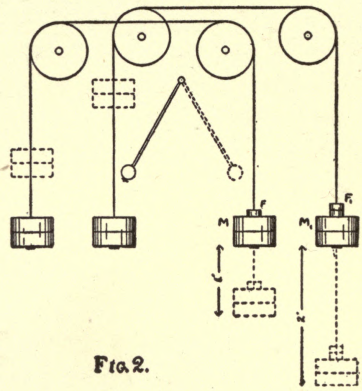

Fig. 2.

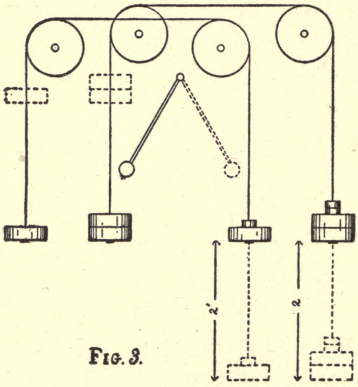

Fig. 3.

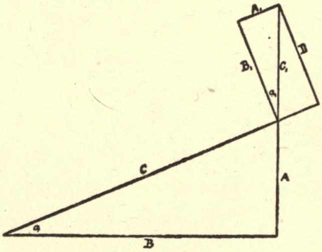

Fig. 5.

Now, I have here two masses (Fig. 3), one twice as great as the other, and if we apply to the larger mass a force twice as great as that which we apply to the smaller mass, we will observe that at the end of 1 second their velocities are the same; or, in other words, for equal velocities the forces must be proportional to the masses.

Or, to generalize: velocities are inversely proportional to masses, and directly proportional to forces.

Or, by assuming the unit of force as that force which will, in unit time, give unit mass unit velocity, we may formularize the phenomena we have observed by writing

$$\text{Force} \times \text{Time} = \text{Mass} \times \text{Velocity}, \quad . \quad . \quad . \quad . \quad (1)$$

from which we may get by transposition

$$\text{Force} = \frac{\text{Mass} \times \text{Velocity}}{\text{Time}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\text{Time} = \frac{\text{Mass} \times \text{Velocity}}{\text{Force}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$\text{Mass} = \frac{\text{Force} \times \text{Time}}{\text{Velocity}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$\text{Velocity} = \frac{\text{Force} \times \text{Time}}{\text{Mass}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The product of "force" into "time" is called "impulse," and the product of "mass" into "velocity" is called "momentum"; the equation teaches us that an "impulse" is equal to the "momentum" which it produces.

It is chiefly with the practical application of the laws we have just enunciated that we have to do in regulating the speed of water-wheels.

Let us examine still further into the matter. We know that if we let any weight fall freely under the force of gravity—or as it is usually written, with a force $=g$,—at the end of one second it will have acquired a velocity of approximately 32.2 ft. per second; or, if we throw any weight up with an initial velocity of 32.2 ft. per second, the force of gravity will stop it in 1 second.

In the latter case, the velocity at the start $=32.2$, and the velocity at the end of the second $=0$, and the mean or average velocity and the distance it will travel $=32.2 \div 2 = 16.1$ in the first second. Therefore the work in foot-pounds which any weight

can do by being thrown vertically with an initial velocity of 32.2 ft. per second $=$ weight $\times \frac{g}{2}$ ft.

Please note that in above case the initial velocity $(V) = g$ or 32.2, and that $V \div g = 1$.

If, however, we have thrown the weight upward with an initial velocity twice as great as before, i.e., 64.4 ft. per second, the force of gravity would stop it in 2 seconds; but the mean velocity in this case is $64.4 \div 2 = 32.2$, which is twice what it was before, and we must also note that it was traveling upward twice as long as before; hence, by doubling both the velocity and the length of time, it will ascend four times as far. Thus, by doubling its $V \div g$, which in the latter case $= 2$, we have enabled the weight to do four times the work. Or, we may truthfully state that in the second case the work equals that in the first case multiplied by $(V \div g)^2$; or, formulating it,

$$\text{Work in foot-pounds} = \text{Weight} \times \frac{g}{2} \times \left(\frac{V}{g}\right)^2, \quad . \quad . \quad (6)$$

$$\text{Work in foot-pounds} = \text{Weight} \times \frac{g}{2} \times \frac{V^2}{g^2}, \quad . \quad . \quad . \quad (7)$$

$$\text{Work in foot-pounds} = \text{Weight} \times \frac{V^2}{2g}; \quad . \quad . \quad . \quad . \quad (8)$$

or we may more conveniently write it

$$\text{Work in foot-pounds} = \frac{\text{Weight}}{g} \times \frac{V^2}{2}, \quad . \quad . \quad . \quad . \quad (9)$$

which is the form in which we will have occasion to most often use it.

As this is a universal law applicable to any force and any velocity, it is applicable to water falling under the influence of gravity.

To fix it in our minds let us apply it numerically to the masses with which we have been experimenting. Start with the masses as shown in Fig. 2.

Let the masses M and M_1 be equal, and let them each be numer-

ically equal to 1. Let $F=2$ and $F_1=4$. Let their time of action $T=1$ second, then their velocities at the end of the time T will be

$$V=\frac{FT}{M}=\frac{2\times1}{1}=2,$$

$$V_1=\frac{F_1T}{M_1}=\frac{4\times1}{1}=4.$$

Now at the end of the time T stop the action of the forces F and F_1 and apply to the masses M and M_1 in an opposite direction a new force $F_2=2$. This new force will stop the masses in the following lengths of time:

$$T_1=\frac{M\times V}{F_2}=\frac{1\times2}{2}=1,$$

$$T_2=\frac{M_1\times V_1}{F_2}=\frac{1\times4}{2}=2.$$

But the space S_1 and S_2 through which they will travel before they stop will be

$$S_1=\frac{VT_1}{2}=\frac{2\times1}{2}=1,$$

$$S_2=\frac{V_2T_2}{2}=\frac{4\times2}{2}=4,$$

or the masses are as. 1:1
and the forces applied to them are as. 2:4
for lengths of time which are as. 1:1
which give them velocities which are as. 2:4
and cause them to travel through spaces which are, during time T, as. 1:2
consequently doing work on them which are as

$$\left(1\times\frac{2\times2}{2}\right):\left(1\times\frac{4\times4}{2}\right) \text{ or as. . . . . . } 2:8$$

They can then oppose equal forces through spaces which are as. 1:4
during lengths of time which are as. 1:2

consequently doing amounts of work which are as

$$FS \text{ and } FS_1 \text{ or } (2\times1):(2\times4) \text{ or} \ldots\ldots\ldots 2:8$$

In the above case the masses M and M_1 should consist of weights of 32.2 lbs.; that is, the weight at each end of the wire should be 16.1. The force $F=2$ lbs., $F_1=4$ lbs., $F_2=2$ lbs. A pendulum 39.1 inches long from center of weight to point of support will beat near enough seconds for ordinary experimental purposes. A few experiments carefully carried out with the apparatus shown in Figs. 1, 2, and 3 will teach one more about the relations of mass, force, time, and velocity than can readily be learned in any other way.

We are now prepared to see the application of the laws, and the formulæ we have written, directly to one of the most important details connected with the installation of water-wheels. This can be best shown by an example.

We have here two water-wheels (Fig. 4) operating under the same head, which we will assume to be nine (9) ft. You will

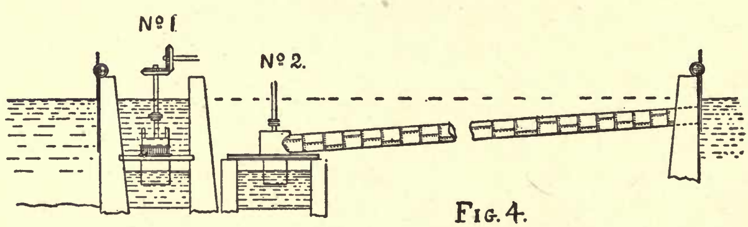

FIG. 4.

observe that although the head is the same in both cases, one wheel—which we will designate No. 1—is set in an open flume of ample size; while the other wheel—which we will designate as No. 2—is in a closed flume connected to open water by a long closed pipe which is nearly horizontal. The behavior of these two wheels when operating under variable load is entirely different.

Let us assume, for the purposes of argument, that the efficiency of the wheel is the same at all stages of gate, and that the amount of water which passes through the wheel is proportional to the gate-opening, and that the power of the wheel is proportional to the amount of water which passes through it under constant

pressure. Now, if the wheel is operating at full gate and half the load is suddenly thrown off, and the suitably designed governor attached to the wheel promptly shuts the gates so that only one half as much water can pass as when the wheel was at full gate, it is evident that the speed will remain comparatively constant.

Let us see if this will be the case with wheel No. 2. If it is operating at full load, and half the load is instantly thrown off, and the governor promptly shuts the gates so that only half as much water can pass, it is evident that the velocity of the water in the closed pipe must be reduced one half.

If we assume that the water in the pipe weighs 1,000,000 lbs., and has a velocity at full head of 4 ft. per second, its energy (see formula 9) $= 1,000,000 \div 32.2 \times 4^2 \div 2 = 248,440$ ft.-lbs., and if the water velocity at half load is 2 ft. per second, then its energy $= 1,000,000 \div 32.2 \times 2^2 \div 2 = 62,110$ ft.-lbs., and the difference between these two amounts of energy, $248,440 - 62,110 = 186,330$ ft.-lbs., must be expended upon the water-wheel before the water velocity is reduced to 2 ft. per second.

If it were expended in one second it would $= 186,330 \div 550$ H.P., but this is a little quicker than we would expect to do in practice. Suppose we slow up the water-column in two seconds, then the energy expended $= 186,330 \div 550 \times 2 = 169$ H.P. for two seconds. The above value of H.P. would not hold strictly true unless the rate at which the gate closed was proportional to the rate at which the water-column slowed up; but the total foot-pounds expended on the wheel would be as above stated. To find the exact value of H.P. at any instant of time would require a more elaborate mathematical treatment of the problem than the time now at our disposal permits; but the significant fact to which I wish to call your attention is that this work done upon the water-wheel in slowing up the water-column is entirely independent of, and in excess of, the work which is expended upon the water-wheel when it is working normally at half gate, with the water-column moving at a fixed velocity.

It is evident that the above amount of work done upon the wheel while the water-column is slowing up would tend to make the speed of the water-wheel run high if the governor only half

closed the gates. In fact, the governor would have to set the gates much nearer closed than one half; or, to speak more accurately, the governor would, at each instant of time, have to hold the gates at such a position that the power developed by the wheel, due to the working-head plus the instantaneous value of power being developed by the slowing water-column, equaled the load upon the wheel.

This might be found to be quite unfeasible, for the pressure developed on the closed pipe and wheel-case might be dangerous, or the gate might be too ponderous or too badly rigged to permit of the requisite promptness of motion.

The maximum pressure which would be developed at any instant of time at the water-wheel would be an impossible thing to calculate without knowing a great deal more about the venting areas and time-ratio of closing them than can ordinarily be found out in practice. All that can be predetermined is what may be called, for want of a better term, the time-average pressure. This can easily be determined as follows:

Let P = the time-average pressure;

L = the length of the closed flume in feet;

V = the water velocity in feet per second;

T = the time in seconds in which the water velocity is arrested;

K = the area of a square inch expressed in square feet = .00694. Then

$$P=\frac{K\times 62.4\times L\times V}{32.2\times T} \quad \ldots\ldots\ldots\ldots \quad (10)$$

It will be observed that

$$\frac{K\times 62.4}{32.2}=.01324 \text{ is a constant, call this } K_1,$$

and the formula becomes

$$P=\frac{K_1\times L\times V}{T} \quad \ldots\ldots\ldots\ldots \quad (11)$$

Applying this to the flume we have been discussing, in which

$$L=300,$$
$$V=2,$$
$$T=2,$$

we have

$$P=\frac{.01324\times 300\times 2}{2}=3.97.$$

As a water-column 1 ft. high exerts a pressure of .43 lb. per square inch, it follows that a pressure of 3.97 lbs. per square inch represents a head of $3.97 \div .43 = 9.2$ ft. In other words, if the pressure on the wheel could have been kept constant all the time the water-column was slowing up from 4 ft. per second to 2 ft. per second, the wheel would have been working under $9+9.2 = 18.2$ ft. of head, instead of under 9 ft. of head as it should have been.

From experience we know that it is impossible to close the water-wheel gates at such a rate as to keep the pressure constant, and as a matter of fact, during some portion of the two seconds the water-pressure would have been greatly in excess of 3.97 lbs. per square inch above normal, with a correspondingly large disturbance of the speed.

We may note a curious fact in this connection. With a water-wheel set like No. 2 in Fig. 4, working at nearly full gate, and if under these conditions a large portion of the load is instantly thrown off and the governor is of unsuitable design and does not compensate for the kinetic energy of the slowing water-column, it may be found by experiment that the speed will run higher than though there were no governor at all. This is for the reason that for an interval of time the wheel is working under light load and a greatly increased head, and there is, consequently, a greatly increased speed; or, we may say that the amount of energy applied to the wheel under the increased pressure, even though the gate areas have been somewhat reduced by the governor, is greater than would have been the case had the gates not been moved at all.

The first remedy which suggests itself is to place large relief-

valves near the wheel-case, so that they will open and let the water escape if the water much exceeds the static head. This would help matters somewhat upon load suddenly going off, but would it help matters upon load suddenly going on? Let us examine this matter.

Suppose the wheel is working at half load with the water-column moving at a rate of 2 ft. per second, and the whole load is instantly thrown on the wheel. The governor will promptly open the gate wide, but the water-wheel cannot develop its whole power until the water-column has attained a velocity of 4 ft. per second. To gain this extra 2 ft. per second the water-column must have expended upon it the same amount of work which it expended in losing its 2 ft. per second, namely, 186,330 ft.-lbs., and this must be deducted from the work the wheel will do normally at full gate; so that the instantaneous value of power developed by the wheel while the water-column is gaining velocity would equal the normal power of the wheel at full gate minus the instantaneous value of power being expended upon the water-column in getting up to speed.

It is evident that the speed of the water-wheel would fall considerably below normal and there would be absolutely no remedy for it in the present state of the art. I say this advisedly and have not forgotten the question of fly-wheels, which is undoubtedly in all of your minds at the present moment.

Let us now consider how long it will take the water-column to get up to speed. First let us look again at wheel No. 1 for a moment. We know that the velocity of water falling without friction may be expressed by the formula.

$$V=\sqrt{2g\times H}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where V = velocity in feet per second,
$g = 32.2$,
H = head in feet.

Taking $2g$ outside of the square-root sign, we have

$$V=8.025\sqrt{H}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

This is the velocity with which water should enter the water-wheel.

For purposes of simplicity I have in this paper ignored the corrections which should be made for water friction on surfaces and in orifices, as they do not alter to any large extent the stubborn facts we are considering. Such corrections are beyond the scope of this paper.

Applying formula No. 13 to wheel No. 1 and assuming that the water enters the wheel without friction, we have

$$\left.\begin{array}{l}\text{Velocity of water entering}\\ \text{wheel under 9 ft. head}\end{array}\right\} = 8.025\sqrt{9} = 24 \text{ ft. per second.}$$

Now, as the time required for a falling body to acquire a given velocity $= \dfrac{V}{g}$, we find in the case of wheel No. 1

$$\left.\begin{array}{l}\text{Time in seconds for water to ac-}\\ \text{quire spouting velocity into}\\ \text{wheel under 9 ft. head}\end{array}\right\} = \frac{V}{g} = \frac{24}{32.2} = .7 \text{ second}$$

Thus, if the gates being closed were instantly opened, the water would be doing its full amount of work on the wheel in seven tenths of a second.—To make the above absolutely true it would be necessary to assume that the water would enter the wheel with equal freedom at all stages of gate, which is not the case, but it is sufficiently near the truth for our present argument. We are also ignoring what is known as the velocity of approach for reasons previously stated.

To make sure that our figures are right let us calculate the value of V from our fundamental equation (No. 5),

$$V = \frac{FT}{M}.$$

Assume a vertical water-column of 1 sq. ft. area and 9 ft. high. Its weight is $62.4 \times 9 = 560.7$: this $= F$. Then

$$V = \frac{561.6 \times .7}{\dfrac{561.6}{32.2}} = 22.6,$$

which is a close approximation to the value of $V=24$ previously found, the slight discrepancy being due to the fact that 62.4 is not the exact weight of a cubic foot of water, and 32.2 is not the exact value of g. It might also be added that the square root of $2g=8.025$, which is usually given in books on hydraulics, and which was previously given in formula No. 13, is a trifle too large, and a closer approximation to truth will be obtained by calling it 8.02. By using better values we can bring out $V=23.8$.

Now, in case of wheel No. 2 the water will behave in an entirely different manner. We know that the spouting velocity at wheel No. 2 is the same as at wheel No. 1 minus the friction of the pipe. Unfortunately, we are concerned not only with the spouting velocity at wheel No. 2, but with the length of time it will take to attain spouting velocity at wheel No. 2. The water, instead of falling vertically as in wheel No. 1, runs down an almost horizontal inclined plane. A large part of the force of gravity is applied perpendicularly to the inclined plane and the small remainder is applied to shove the water down the inclined plane. A diagram will make this clear.

Let A in Fig. 5 = the head in feet from open water above the entrance to the flume to tailwater level.

Let B = the horizontal distance in feet from entrance to flume to draft-tube at tailwater level.

Then C = the hydraulic slope.

Project $C_1=A$ = the hydrostatic head.

Draw D perpendicular to C.

Complete the parallelogram.

Then the force C_1 (which we must remember is the hydrostatic head) is equal to the forces A_1 and D, of which the latter is wholly sustained by the reaction of the plane C_1, while A is wholly effective in accelerating the motion of water down the slope C. We evidently wish to know the value of A_1 in terms of the triangle ABC, which is similar to the triangle $A_1B_1C_1$.

We know that $A_1=C_1$ sine a_1. Then as $C_1=A$ and $a_1=a$, it follows that $A_1=A$ sine a. Let us designate the value of A_1 so found as f.

We have seen (Fig. 3) that the time to give equal masses equal velocities is inversely proportional to the forces.

Calling T the time to acquire spouting velocity down A, and T_1 the time to acquire spouting velocity down C, we may write

$$T_1 : T = f : A, \qquad (14)$$

from which we get

$$T_1 = \frac{TA}{f}. \qquad (15)$$

Let us apply this equation (15) to the case of wheel No. 2 in Fig. 4.

The head $A = 9$ ft.

Assume $B = 298.7$ ft.

Then $C = \sqrt{9^2 \times 298.7^2} = 300$ feet,

$$\text{Sine } a = \frac{A}{C} = \frac{9}{300} = .03,$$

$$A_1 = A \text{ sine } a = 9 \times .03 = .27 = f,$$

$$A = 9,$$

$$T = .7, \text{ previously found;}$$

then

$$T_1 = \frac{TA}{f} = \frac{.7 \times 9}{.27} = 23.3$$

seconds required for water to acquire spouting velocity down spouting slope C.

It may be noted that

$$T_1 = \frac{TA}{f}$$

may be written

$$T_1 = \frac{TA}{A \text{ sine } a}, \qquad (16)$$

which by cancellation becomes

$$T_1 = \frac{T}{\text{sine } a}. \qquad (17)$$

which is its simplest form, and in which it should be used.

To make the above reasoning plainer, I have plotted (see Fig. 6) lines showing the time necessary for water to acquire spouting velocities into the two water-wheels in Fig. 4. Line 0-1 shows the time for water to acquire any velocity up to spouting velocity into wheel No. 1; line 0-2 shows the time for water to acquire any velocity up to spouting velocity into wheel No. 2.

It naturally occurs to one in this connection that the water never has occasion to acquire spouting velocity in the flume of wheel No. 2; in fact, we assumed that the maximum water velocity in this flume was only 4 ft. per second, which is only one sixth of

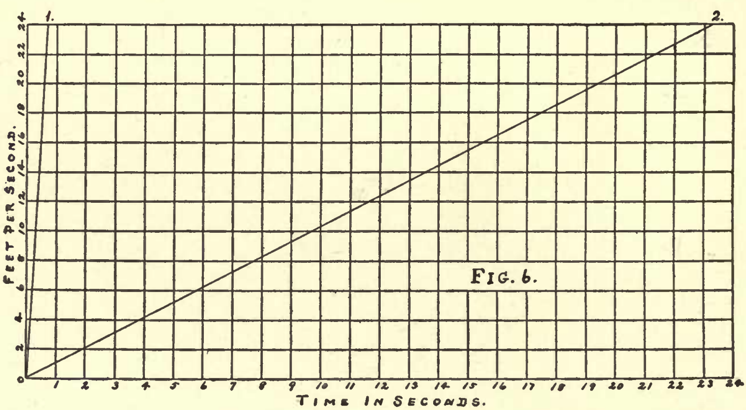

FIG. 6.

spouting velocity. It can be shown mathematically, and experiment proves, that this does not interfere with the line of reasoning we have been following. If, instead of the end of the flume being wide open, it were five sixths closed, the remaining sixth being an orifice (the venting areas of the water-wheel) capable of being varied at will, it would simply mean that in the flume the value of $g=32.2$ would be considerably reduced. This new value we should calculate and call it G. We could then substitute it for the value of g we have been using in our calculations, and the ratio of velocity and time in the open flume of wheel No. 1 and in the closed flume of wheel No. 2 would be found to be the same that we have already ascertained.

But it may be argued that we are concerned with the force which the water will apply to wheels No. 1 and No. 2 while the

water is getting up to spouting velocity, and not with the velocity of the water itself. Let us see at what rate the water will develop its full amount of energy on the two wheels we have been considering.

The theoretical amount of energy which flowing water can apply to an obstacle, advantageously placed in its path, may be expressed as follows:

$$P=F+F=2W\frac{V}{g}=4wa\frac{V}{2g}, \quad . \quad . \quad . \quad . \quad . \quad (18)$$

where $P=$ the theoretical energy developed,

$F=$ the force of impulse and also of reaction,

$W=$ weight of water flowing per second,

$w=$ weight of 1 cu. ft. of water,

$a=$ cross-section of the stream in square feet.

It will be noted that if we assume $a=1$, we may regard

$$\frac{4wa}{2g}=3.9$$

as a constant, which, multiplied by the square of the velocity of flow in feet per second, will give the theoretical force which the water will develop. I have calculated the force developed at wheel No. 1 and No. 2, as shown in Fig. 4, for each tenth of a second, beginning with the water at a rest in both cases and ending with spouting velocity, and have plotted the values in Fig. 7. Curve 0-1 shows the rate at which water standing at a rest develops its full energy on wheel No. 1; curve 0-2 shows the rate at which water standing at a rest develops its full energy on wheel No. 2. You will note in the case of wheel No. 1 how very promptly the energy gets to its maximum value, and in the case of wheel No 2 how the energy lags for a considerable time before it arrives at anything like its maximum value.

I wish to emphasize this line of reasoning, because it is perhaps the most important thing to be considered in setting water-wheels where speed regulation is a desideratum. We can, in an imperfect way, provide for the expenditure of the energy necessary to slow up a water-column, but there is no way to make a water-column,

while gaining velocity, do the work it is capable of when it has arrived at full velocity.

The important fact to which I want to especially call your attention is that the difficulty is measured not only by the length of the closed flume, but is inversely proportional to the sine of the angle of hydraulic slope. When the sine becomes 1; that is, when the angle is 90°,—or, in other words, when the closed flume

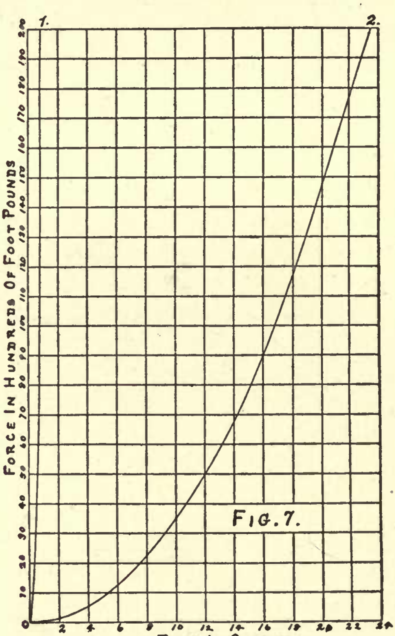

Fig. 7.

is vertical,—then the difficulties due to the fact that water moves slowly under the influence of gravity have reached their minimum and the speed regulation will be the best obtainable. As the sine of the angle of hydraulic slope grows less, then the obtainable regulation grows worse.

There is one way in which the difficulties attendant upon a small angle of hydraulic slope may be in a measure compensated for, and that is by means of a stand-pipe.

In an electric plant it is not usually of such importance that

a load change amounting to the full capacity of the wheels be followed by a small speed variation, as that the comparatively large loads which go off and on for short intervals of time shall not disturb the speed to any great extent. Here is where the stand-pipe is of value. If a portion of the load goes off instantly and the correctly designed governor promptly closes the gates to the correct position, the excess of water will flow out over the top of the stand-pipe and the water velocity in the flume will not be arrested so promptly as though there were no stand-pipe; neither will the pressure at the wheel be much increased. To obtain these results, the stand-pipe should be only a very little higher than the water level in the pond. It should be located as near the wheels as possible, and its top should be turned over so that the escaping water can be led to some convenient point of discharge.

If, after a load has gone off instantly, it comes on again in a short interval of time, it finds the water velocity in the flume but little diminished, and also the vertical water-column in the stand-pipe is ready to apply its energy to the water-wheel in the most advantageous manner. To make the last factor of much practical use, the cross-section of the stand-pipe must be sufficiently large to prevent the level of the enclosed water-column from falling much while the water in the closed flume is gaining its lost velocity. As a general statement, the larger the diameter of the stand-pipe and the less its height above the hydrostatic level, the better will be the speed regulation. There has not, as yet, been sufficient practical experience with stand-pipes to formulate rules which will solve the least diameter which will result in any desired degree of speed regulation.

In the writer's experience, it has been found that the use of a stand-pipe of ample proportions will render a plant governable within very close limits under ordinary operative conditions which had proved to be utterly ungovernable before the stand-pipe was installed.

From what has been said above it will be seen that a stand-pipe is chiefly of use in aiding a good governor to maintain a comparatively constant speed under those frequently recurring load changes which obtain in electric plants,—especially in electric

railway and power plants. It also gives perfect protection against dangerous water-pressures being developed when circuit-breakers open, or when, for accidental reasons, it is necessary to shut down the water-wheels instantly.

A stand-pipe will not—unless of very large diameter—enable a good governor to maintain a good degree of speed regulation if the load be increased from friction load to full load instantly, where the angle of hydraulic slope is small, unless such increase of load takes place before the water in the enclosed flume has lost much of its velocity.

The writer had intended to refrain entirely from submitting designs showing the proper setting of water-wheels, for the reason that such a large number of typical plans would be required to cover all probable cases that it would be hopeless to treat the subject properly without extending this paper beyond proper limits. He cannot, however, resist the temptation to introduce at this point one design (shown in Fig. 8) which has proved to be singularly adapted to the demands of water-power-driven electric plants.

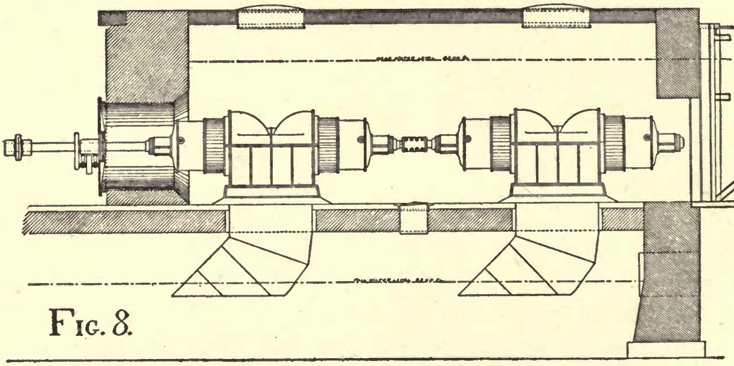

FIG. 8.

It will be noted that the wheels are arranged for direct connection. The angle of hydraulic slope is practically 90°, giving the best possible conditions for speed regulation. The governor may be placed directly outside the flume-head and connected to the gates in the simplest possible manner. The speed variations of the main shaft may be transmitted to the governor by one belt. The requisite R.P.M. may be obtained by varying the diameter

and number of wheels. The number or size of wheels on the one shaft may vary from one to as many as can be handled by one governor, or as may be required by the capacity of the electrical unit. This general design has found favor in a number of the most prominent plants in this country as well as in Europe. The regulation is invariably good if a suitable governor be used.

It is usually the case that part of the head utilized in modern plants is below the water-wheel in the shape of a draft-tube; in fact, where horizontal wheels are used, it is practically necessary to have them a number of feet above tailwater level for convenience of connection to the driven machinery.

The same general rule holds good in regard to draft-tubes, which, we have found, applies to closed flumes. They should be as short and as nearly vertical as possible. The maximum vertical length of a draft-tube is, of course, limited by the atmospheric pressure. The water stands in the draft-tube for the same reason that mercury stands in a barometer. The specific gravity of mercury is 13.6: that is, it is 13.6 times as heavy as water. Atmospheric pressure holds mercury up in the barometer tube,—let us say 30 ins. or $2\frac{1}{2}$ ft.,—therefore it will hold water up in the draft-tube $2.5\times 13.6=34$ ft.; that is, it would do so if the draft-tube were air-tight. The external atmospheric pressure at the top of such a draft-tube would be 14.7 lbs. per square inch. There are few draft-tubes that would stand that pressure without leaking air. This fact is well recognized by hydraulic engineers, and it is rare to find draft-tubes 25 ft. high from tailwater level to water-wheel centers. If the water-wheel is likely to be subjected to large load variations, it is very desirable that the draft-tube should have a much less vertical height for the following reason:

At the bottom of a 25-ft. vertical draft-tube the atmospheric pressure is forcing the water up with a pressure of 14.7 lbs. per square inch, and the weight of the water is pressing down with a pressure of 10.81 lbs. per square inch: that is. the difference between the air-pressure and the weight is $14.7-10.81=3.89$ lbs. per square inch. Now, if the water velocity in the draft-tube is suddenly arrested by shutting the water-wheel gates, the kinetic energy of the slowing water-column will be found in the down-

ward momentum of the water. This may easily create a downward pressure greater than 3.89 lbs. per square inch, in which case a vacuum would be formed in the upper part of the draft-tube and the column of water would sink in the draft-tube and immediately after would rush upward again, striking the bottom of the wheel with great violence. If we were so fortunate as to escape an accident of the kind above described we should find that with a draft-tube of considerable height there is a tendency for air to leak in, and this, under the negative pressure of the weight of the water, expands into a partial vacuum so that the draft-tube will be only partly filled with water, and as the position of the water-wheel gates varies as the load changes, the water-column in the draft-tube will sway up and down, producing the effect of a pulsating head on the water-wheel. This is very detrimental to good speed regulation, and is a very common annoyance encountered in practice. The performance of such a draft-tube may be easily illustrated by holding a mercurial barometer in the hand and slowly moving it up and down.

Air-chambers on flumes, to give protection against water-hammer effects, are of very little practical use unless of ample size, even if they are full of air. The writer examined a plant so located that the bursting of the flume would have destroyed the whole plant and ruined an investment of at least $100,000. At the lower end of the flume was a large air-chamber. The superintendent in charge pointed with pride to it, and confidently expressed the belief that it afforded ample protection against the dangerous strains on the flume due to water-hammer. Upon examining the air-chamber it was found to be entirely filled with water, and it had probably been in that condition for a considerable length of time. Water under pressure absorbs air with great facility. An air-chamber should be provided with an air-pump which may be readily connected to some convenient source of power, and with a gauge-glass to show the water level. When so arranged, and if of ample size, it affords considerable safety against pressure developed when load goes off suddenly; but it is of no practical use as an aid to the governor in maintaining constant speed.

Aside from designing the water-column along the lines already

suggested, so that the water may gain its working velocity in the least possible time and also so that it may add to or take from the water-wheel the least amount of the kinetic energy of the water, the next most important thing is the design of the water-wheel gates and the method of connecting them to the governor.

As has already been pointed out, the gates are of necessity large and heavy, and yet they must be moved with great promptness and precision. The writer has had occasion to investigate with more or less accuracy the number of foot-pounds necessary to open and close the gates of several hundred water-wheels, and the surprisingly large variation in the amount of energy required leads him inevitably to the conclusion that this matter has not received in many cases the careful engineering treatment which it deserves.

Water-wheels are of many designs and sizes, and work under many different conditions of head, but there would seem to be no adequate reason why the gate of one water-wheel developing a certain amount of power under a given head should require only 1000 ft.-lbs. to completely open it, and the gate of another water-wheel of different make, developing the same amount of power under the same head, should require 60,000 ft.-lbs. Yet such has been found to be the case. The above example, taken from actual practice, is by no means unusual; and scores of such cases could be cited showing relatively absurd figures.

Some builders prefer to use cylinder gates on their wheels; others prefer wicket gates; while still others adhere to register gates. It is not the intention of this paper to enter into a critical comparison of the merits of these various types of gate, and in fact, from the standpoint of speed regulation, no such comparison is necessary for the good and sufficient reason that there are wheels on the market of all three of the above kinds which show little to be desired in the ease with which the gates may be moved. It is also true that there are makes of wheels of all three kinds which cannot be governed accurately under variable loads, simply for the reason that their gates cannot be moved quickly enough.

It is often necessary to start a gate from a rest and completely open or close it in 2 or 3 seconds, or give it a proportionately smaller motion in a proportionately shorter space of time. Or,

what is still more severe, it is often necessary that while a gate is opening or closing, its motion be instantly stopped and reversed.

If one will watch a thoroughly first-class governor handling the gates of a water-wheel which is driving an electric generator operating on a variable load, one is convinced of the fact that the governor has to develop considerable amounts of energy in surprisingly short spaces of time, and that the rigging connecting the governor and the gates is subjected necessarily to considerable strain, from which it follows that the easier the gates move the less chance there is of stripping gears and twisting off shafts, to say nothing of relieving the governor itself of unnecessary strain.

All gears between governor and gate—except immersed racks and pinions—should be cut, of first-class workmanship, and not too large for the work required of them. The latter precaution is necessary to prevent the MV^2 energy in the gears themselves destroying the rigging when the direction of motion is suddenly reversed. Shafts should be of just sufficient size to give an ample factor of safety, and prevent torsional difficulties, for it is absolutely necessary that the smallest amount of motion of the governor shall be transmitted accurately to the water-wheel gate. Lost motion in gears and twisting of shafts are fatal to good regulation. Hand-wheels should be so arranged that they may be entirely thrown out of connection with the rigging while the governor is in action, or they may be unkeyed in some simple manner.

Counterbalancing a gate is not the equivalent of having it in water balance. All vertical cylinder gates are necessarily out of balance to an amount equal to their immersed weight, but that isusually so small that it is not necessary to counteract it with a counterweight.

Some designs of gate show a violent tendency to close or stay closed. It is the custom to counterbalance such gates, and this practice leads to endless trouble on account of the kinetic energy in the counterweight. It being often necessary to reverse the motion of the counterweight suddenly, the kinetic energy expended at the moment of reversal is often sufficient to wreck the rigging. If counterweights must be used, it should be remembered that their kinetic energy is proportional to their weight, but also proportional to the square of their velocities; from which it follows that a heavy, slow-moving weight does less damage than a light, rapid-moving one.

Some general statements may be made in regard to the design of water-wheel gates adapted to plants in which it is desirable to obtain good speed regulation.

It has been the custom of late to cast onto cylinder gates fingers reaching out between the guides. These innocent-looking devices, which are supposed to guide the water into the wheel properly, and hence raise its efficiency, are a source of no end of trouble when it comes to moving the gate quickly enough to produce good speed regulation. The direction of motion of the water as it enters the wheel is always such that it presses these fingers downward with tremendous force, giving the gate a strong tendency to close. By removing these fingers, the amount of energy necessary to open the gates can always be reduced by at least one-half, and oftentimes more than that. There are scores of water-wheels on record which were so much out of balance, due to the fingers on the gates, that it was found impracticable to govern them satisfactorily on account of gears stripping and shafts twisting off. In the writer's experience it has always been found practicable to govern these wheels by removing the fingers.

Now as to the question of efficiency. The writer has often had to meet the argument of the few per cent of efficiency supposed to be lost by removing these fingers, and to answer this question tests have been made which show that there is no material gain in the efficiency of a water-wheel set under ordinary working conditions by attaching fingers to the gate.

Two vertical cylinder-gate wheels of the same size and make were set in open flumes side by side. The head was precisely the same in both cases. Both wheels drove electric generators of the same make, type, and size. Both wheels were furnished by the maker with cylinder gates precisely like and provided with fingers. It was found impracticable to govern these wheels properly, on account of the gates working so hard and being so much out of balance. The fingers were removed from the gate of one wheel, and it was at once found that the wheel governed very satisfactorily under a very variable load. Then a test was made of the efficiency of the two wheels, one with and one without fingers on the gates. Wires were brought up from the gates, carried over pulleys, kept taut by small weights, and they terminated in pointers reading on

the same scale. A constant electrical load was switched onto one generator, and the position of the pointer indicating gate position was noted. Then the load was switched onto the other generator (the speed being kept the same in both cases), and it was noted that the pointer of the second unit stood at the same point at which the pointer of the first unit had previously stood. This experiment was repeatedly tried at a number of different loads, from slightly above friction load to nearly the full capacity of the wheels. So far as could be observed, the efficiency of the wheel without fingers on the gate was as good as that of the wheel with the fingers. The particular test above described was made by Mr. J. H. Wilson, in the plant of the Berlin Mills Co., Berlin Mills, N. H.

The writer is aware that this test is not the equivalent of a Holyoke test, but it is certainly of great interest to the practical engineer who is harassed by the thought that in avoiding the Scylla of bad efficiency he will surely be wrecked on the Charybdis of an ungovernable gate.

There are a number of other details of cylinder-gate construction which time and space will not permit us to touch upon here, but which should be considered by the thoughtful engineer before making a selection. The thing to be borne in mind is that the cylinder and its connections should be of such design that they may be easily moved, and will not bind and run hard in any portion of their travel.

Wicket gates also have their pecularities. Some makers hang them in such a manner that they are practically in water balance in any position, and may be readily opened and closed with a small expenditure of energy. Such gates leave little to be desired, and wheels fitted with gates so designed may be governed with the greatest degree of exactness and without fear of injury to the rigging or governor. The writer has observed, however, that some wicket gates which move very easily have so much lost motion that in certain portions they tend to flop (no other word conveys the idea) first in one direction and then in the other, causing a pulsating speed which is very annoying and apparently inexplicable until one has investigated the cause. The danger of lost motion is greater with wicket than with cylinder gates, but with proper construction it is found in practice that lost motion may be entirely eliminated from wicket gates.

In some wicket-gate wheels the wickets are hinged at one end and attached by the other end by tangential arms to a banjo, which in turn is geared to the shaft going to the governor. Such gates are entirely out of water balance when partly closed, and the more they are closed the more they are out of balance. Wheels with gates of this description are very difficult to govern. Frequently the strength of the wickets and radial arms is not sufficient to withstand the water-pressure, even if sufficient energy can be supplied to them. In recent practice a wheel of this description was found to require some 40,000 ft.-lbs. to open it. Another wicket-gate wheel of different make but the same rated horse-power was found to require only 5000 ft.-lbs. to open it. As another recent instance, it was found that a pair of wicket-gate wheels of the kind described above required 19,000 ft.-lbs. to open them; another pair of different make but the same rated horse-power required but 2500 ft.-lbs. to open them. The wheels compared above were working under the same head.

The way a maker proposes to rig his gate is a good indication of the amount of energy he thinks it will take to move it. If he thinks it is necessary to use worm-gears or multiplying-gears giving a large number of turns to the hand-wheel, it is safe to conclude that in his opinion—and he certainly ought to know—the gate will move hard or be much out of balance. Such wheels it is safe to leave alone if accurate speed regulation under variable load is the end in view.

All practical engineering is a compromise between the desire of the engineer on the one hand to produce a perfect piece of engineering and the unwillingness of the stockholders on the other hand to invest money which will not bring direct returns in the shape of dividends, or, to state the matter more conservatively, there is always a point in each plant beyond which investment must not go, and this point is different for each plant, being fixed by the economic conditions which surround the particular enterprise.

For the above reasons it is impossible to lay down hard and fast rules for the development of water-powers. Assuming that the value of all engineering is measured by the dividends earned, what would be good engineering in one case is bad engineering in another. Yet it is equally true that in an electric plant driven

by water-power the worst possible place to economize is in the channels and conduits which bring the water to and away from the water-wheels, and in the water-wheels themselves and their governors. Dollars saved here are apt to be expensive economy.

The infinite variety in which natural water-powers present themselves makes this a difficult branch of engineering, and much is yet to be learned about it. Yet it is safe to say that enough is now known about the subject to permit almost any naturally good water-power to be so developed and utilized that the plant driven by it may be as readily controlled and its speed maintained as constant as though the driven load were carried by first-class steam-engines. It should, however, be remembered that, no matter how well the plant is designed, good speed regulation cannot be obtained unless the water-wheel governors are of correct design and well built. The value of a governor is measured by two things—the promptness and ease with which it will move the water-wheel gates to the correct position and stop them when they get there, and its ability to compensate by adjustment of the gate for the kinetic energy in the moving water-column. One might add sensitiveness as a mark of a good governor, but nearly all modern governors are very sensitive, though most of them sadly lack the other two qualifications named above. The best made governors on the market will begin to move the water-wheel gates before any tachometer on the market will show a change in speed, yet they would not govern if they did not know when to stop as well as when to start.

I have been requested by a number of gentlemen to say something about fly-wheels. We may approach this subject in the following manner:

When we wish to find out how much energy is stored in a revolving fly-wheel we begin by finding its moment of inertia, which is its weight multiplied by the square of its radius of gyration.

Let J = radius of gyration in feet;

W = weight in pounds;

I = moment of inertia.

$$I = WJ^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The energy in foot-pounds stored in the revolving wheel is as follows:

Let $\mathcal{E}$ = energy stored in the wheel;

$$a = \text{angular velocity in radians per second} = \frac{2n\pi}{60}, \quad . \quad . \quad . \quad (20)$$

where n = revolutions per minute
and $\pi = 3.14159$. Then

$$\mathcal{E} = \frac{Ia^2}{2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Substituting the value of a we get

$$\mathcal{E} = \frac{I\left(\frac{2n\pi}{60}\right)^2}{2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Evolving which we get

$$\mathcal{E} = \frac{In^2\pi^2}{1800}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

which is the form in which the formula is ordinarily used.

It may be simplified, for

$$\frac{\pi^2}{1800} = .00551 \text{ is a constant.}$$

Substituting this we get

$$\mathcal{E} = In^2 \times .00551. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

But you will note that this expression may be conveniently divided into two parts, as follows:

$$\mathcal{E} = n^2 \times (I \times .00551), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

which is equivalent to saying that every fly-wheel possesses a certain quantity which, multiplied by the square of its revolutions per minute, equals the foot-pounds of energy stored in it.

This quantity is its $(I \times .00551)$, which we may symbolize by $\mathfrak{M}$ and we may write

$$\mathcal{E} = n^2 \mathfrak{M}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

Or, substituting our value of I, we may write

$$\mathcal{E} = n^2 (W \times J^2 \times .00551). \quad . \quad . \quad . \quad . \quad . \quad (27)$$

It is evident that this value of $\mathfrak{M}$, which is the energy stored in the wheel when making one revolution per minute, when once found for any particular fly-wheel may be used at once to calculate its energy at any speed, simply by multiplying it by the square of its revolutions per minute.

This value of $\mathfrak{M}$ for the rim of any fly-wheel may be found as follows:

Let w = weight in pounds of 1 cu. ft. of the metal of which it is made;
d = outside diameter of rim in feet;
d_1 = inside diameter of rim in feet;
l = face of rim in feet. Then

$$\mathfrak{M} = \frac{wl(d^4 - d_1^4)}{59{,}814}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

But a close enough approximation to the $\mathfrak{M}$ of a cast-iron fly-wheel of usual shape with light arms may be found as follows:

Let W = weight of wheel in pounds;
d = mean diameter of rim. Then

$$\mathfrak{M} = \frac{Wd^2}{23{,}000}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

Now the question is how much of a fly-wheel do we require in any particular case. Let us return to wheel No. 1 in Fig. 4.

We remember that when the gate of this wheel was suddenly opened wide it was .7 second before the water was doing its full

amount of work on the runner. We must also remember that the curve 0–1 (see Fig. 7) with which the water got up to full power was a parabola. The area outside the parabolic line was the work which the water did do in this .7 second, and the area inside this parabolic line was the work which the water failed to do in the same time. From the law of areas of parabolas it follows that the area inside this curve (which is a half-parabola) is two-thirds of the area of the rectangle enclosing the curve. Or we may say more simply that while the wheel was getting up to speed it performed one-third of the work which it would have done in the same time had it been working at full gate and full speed.

Let us begin to apply this to the design of a suitable fly-wheel. Assume, for simplicity of calculation, that the water-wheel is about 48 ins. in diameter and at full gate develops 100 H.P. and runs at 75 R.P.M. Let us also assume that it must not, upon the whole load being instantly thrown on or off, run more than 4% below or above normal. Its minimum speed must be then

$$75-\frac{75\times4}{100}=72 \text{ R.P.M.},$$

and its maximum speed must be

$$75+\frac{75\times4}{100}=78 \text{ R.P.M.}$$

We must also remember that it was found that we could not completely open the gates in less than 2 seconds. Therefore, supposing that the rate of opening the gate was uniform, the average gate-opening during the 2 seconds was only one-half, and if the wheel during every instant of time had been developing the full power due to the instantaneous value of gate-opening, it would have developed only one-half as many foot-pounds as though it had been at full gate for 2 seconds. But we found that the power lagged .7 second behind the gate-opening, during which time it developed only one-third of the power due to the gate-opening; hence for 2 seconds the wheel developed $\frac{1}{3}$ of $\frac{1}{2}$ the power it would have developed during the same time at full gate, and for

.7 second more it developed one-third of full power. Reducing this to foot-pounds, we have

$$\frac{100\times550\times2}{6}=18{,}333 \text{ ft.-lbs.,}$$

$$\frac{100\times550\times.7}{3}=\underline{12{,}833} \quad \text{''}$$

Adding these we get 31,166 ft.-lbs., which is the total energy developed by the water-wheel in 2.7 seconds.

If working at full gate and maximum flume velocity for that length of time it would have developed

$$100\times550\times2.7=110{,}000 \text{ ft.-lbs.}$$

Subtracting from this $\underline{31{,}166}$ "

we get 78,834 ft.-lbs., which is the amount of energy which must be developed by the fly-wheel before its speed is reduced to 72 R.P.M.

Let us assume that the fly-wheel makes the same number of revolutions as the water-wheel.

Its energy at 75 R.P.M. is

$$\mathfrak{M}\times75^2=\mathfrak{M}\times5625$$

Its energy at 72 R.P.M. is

$$\mathfrak{M}\times72^2=\mathfrak{M}\times\underline{5184}$$

Subtracting one from the other we get 441

The value of $\mathfrak{M}$ is therefore

$$\mathfrak{M}=\frac{78{,}834}{441}=178.$$

From the above value of $\mathfrak{M}$ should be deducted the $\mathfrak{M}$ of the water-wheel itself and the other rotating parts, such as pulleys and armatures. For simplicity of calculation I shall not make this deduction in this case, and we will proceed to design a fly-

wheel of suitable proportions, which shall have an $\mathfrak{M}$ value numerically equal to 178.

As it is to be of cast-iron we will limit its peripheral speed to 65 ft. per second.

Let d= its outside diameter in feet;
p= its peripheral speed in feet per second;
R= revolutions per minute. Then

$$d=\frac{p\times 60}{R\pi}.$$

Applying numerical values we get

$$d=\frac{65\times 60}{78\times 3.14}15.5+.$$

Assume d_1 or the diameter inside the rim = 14.5 ft.

Transpose formula No. 28 so as to get l or the face of the wheel in feet.

$$l=\frac{\mathfrak{M}\times 59{,}814}{w\times(d^4-d_1{}^4)}. \quad \ldots\ldots\ldots \quad (30)$$

Applying numerical values we get

$$l=\frac{178\times 59{,}814}{450\times(15.5^4-14.5^4)}=1.6 \text{ ft.}=1 \text{ ft. } 7 \text{ in., nearly.}$$

Our fly-wheel rim is, therefore, 15 ft. 6 ins. outside diameter, 6 ins. thick, and 1 ft. 7 ins. wide on the face, and would weigh 16,992 lbs.

This strikes one as a very large fly-wheel for a 100-H.P. unit, but it must be remembered that it is intended to perform very severe duty. Moreover, no allowance has been made for the kinetic energy in the fly-wheel arms, nor in the water-wheel itself and other rotating parts. All of these corrections should be made and the proper deduction made from the fly-wheel above designed.

Another correction should also be made which would still further reduce the size of the fly-wheel.

It was assumed that the water-wheel was working at normal

speed and friction load when the whole load was thrown on. Friction load is a part of whole load, and hence when we say we throw on whole load, we really mean that we are throwing on something less than 100 H.P. Also, at friction load the water in the flume had some velocity, and hence we did not have to start the water from a condition of rest, but from a condition of slow velocity.

To make all these corrections involves a considerable knowledge of hydraulics and mechanics. To treat this subject in a complete manner would involve a good many figures, and would extend this paper far beyond proper limits.

It may be noted here that if it is found desirable to change the value of $\mathfrak{M}$ of a fly-wheel after it is designed, it is not necessary to redesign the wheel. The dimensions of fly-wheels are as the fifth roots of their $\mathfrak{M}$'s. We have found a fly-wheel whose $\mathfrak{M}=178$. If we now wish to reduce its $\mathfrak{M}$ to 150, we write

$$\sqrt[5]{178} : \sqrt[5]{150} = 15.5 : \text{diameter}.$$

From which we get

$$\text{diameter} = \frac{\sqrt[5]{150} \times 15.5}{\sqrt[5]{178}} = 15.0 \text{ ft.},$$

or formularizing it we get

$$D_1 = \frac{\sqrt[5]{\mathfrak{M}_1} \times D}{\sqrt[5]{\mathfrak{M}}}.$$

where D = diameter of wheel having given $\mathfrak{M}$,
D_1 = diameter of required wheel,
$\mathfrak{M}$ = given value of $\mathfrak{M}$,
$\mathfrak{M}_1$ = required value of $\mathfrak{M}$.

All of the linear dimensions should be treated in the same way. For example: if we have a design of a fly-wheel drawn to a scale of 1 in. to the foot, and in building the wheel we read the drawing as though it were $\frac{1}{2}$ in. to the foot, then the $\mathfrak{M}$ of the fly-wheel will be $2^5 = 32$ times as large as it would have been if the wheel had been built according to the scale of 1 in. to the foot.

Where we have a fly-wheel of a given $\mathfrak{M}$ and we know how many foot-pounds it will be required to give up or absorb, as the case may be, we may find the resulting speed by the following formula:

$$r=\sqrt{\frac{\mathfrak{E}-\mathfrak{E}_1}{\mathfrak{M}}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

where r = the final revolutions per minute,

$\mathfrak{E}$ = energy in foot-pounds stored in wheel at normal speed,

$\mathfrak{E}_1$ = energy in foot-pounds required of the wheel,

$\mathfrak{M}$ = energy in wheel when making one revolution per minute.

The above formula may be more conveniently written as follows:

$$r=\sqrt{\frac{(\mathfrak{M}\times R^2)-\mathfrak{E}_1}{\mathfrak{M}}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

where R = normal revolutions per minute. Applying to the wheel we have been discussing we have

$$r=\sqrt{\frac{(178\times 75^2)-78{,}834}{178}}=72,$$

which is the minimum revolutions per minute which we first agreed upon.

We have seen that to make even an approximate design of fly-wheel we have required considerable data to work from, and have been obliged to do quite a little figuring, and yet the water-wheel in question was set in the simplest possible manner. Had it been set in a closed flume like wheel No. 2 (Fig. 4), the problem would have been greatly complicated.

In view of these facts, it becomes quite amusing to note the alleged accuracy with which statements are often made in regard to the exact amount of fly-wheel which is required to give stated degrees of speed regulation upon 10%, 25%, 50%, etc., of the load being instantly thrown off or on, when it is perfectly evident that only part of the data is available which would enable only approximate figures to be made.

The one concluding crumb of comfort which the writer is able to offer is found in the fact that in a very large practice he has never found it necessary, with a water-wheel set in an open flume, to install a fly-wheel in order to obtain a perfectly satisfactory speed regulation under any operating conditions of an electric plant. The various rotating parts of the plant, such as water-wheels, armatures, pulleys, etc., having sufficient moment of inertia and angular velocity to enable a first-class governor to hold the speed within very satisfactory limits under any sudden load changes which occur in the actual operation of the plant.

If the design of the hydraulic part of the plant is bad, it is wiser to try and improve it rather than lean largely on fly-wheel effect, which is a weak prop at the best.

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